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DEVELOPMENT OF A DIAGNOSTIC SYSTEM FOR OFF-HIGHWAY VEHICLE POWE--ETC(U)
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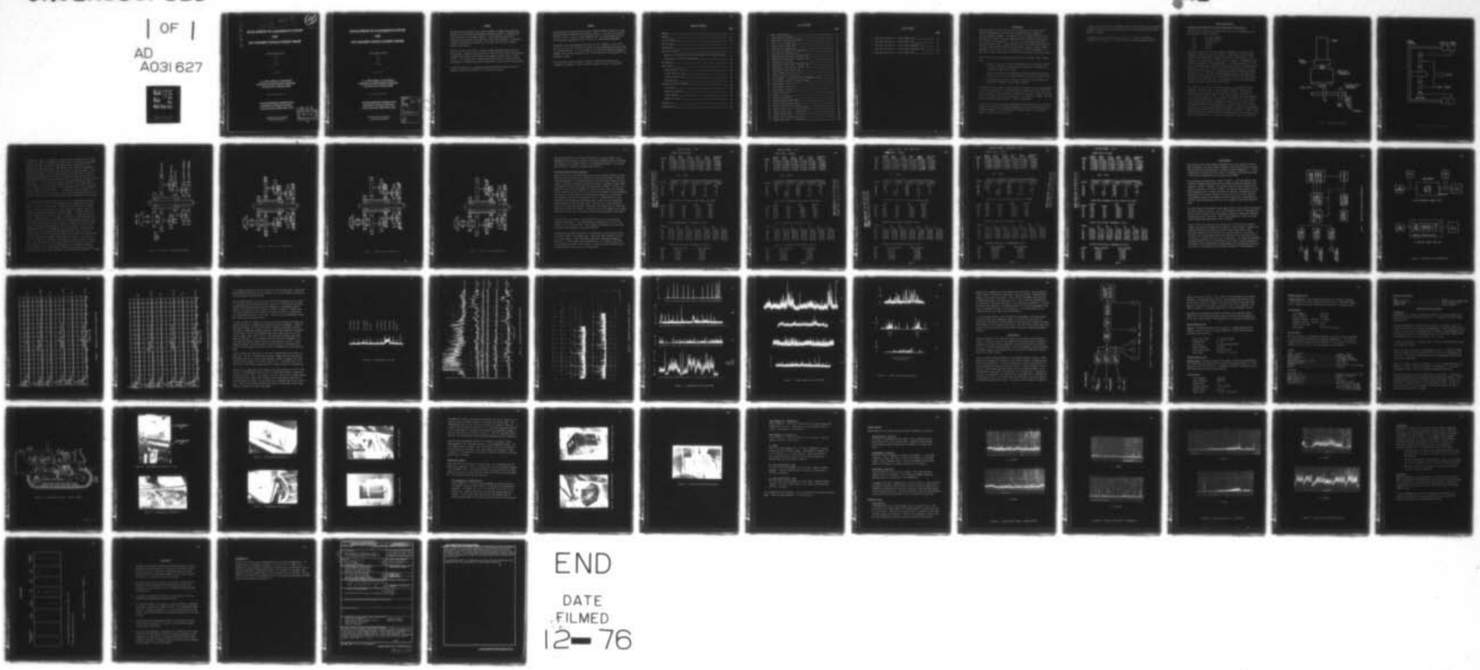

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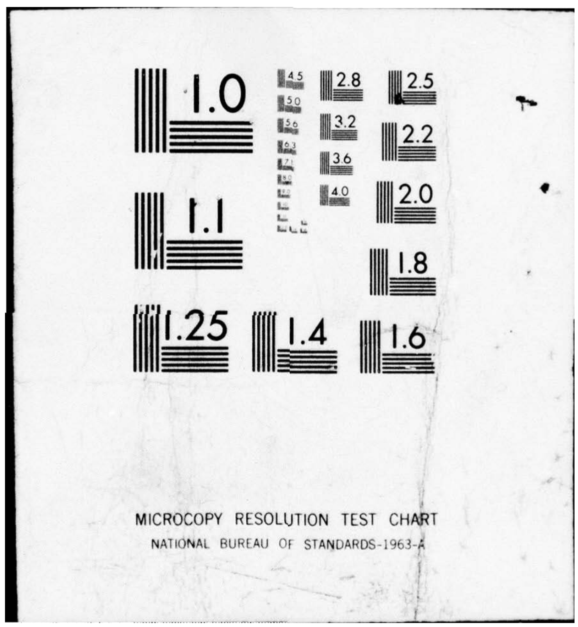
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DEVELOPMENT OF A DIAGNOSTIC SYSTEM FOR OFF-HIGHWAY VEHICLE POWER TRAINS

FINAL TECHNICAL REPORT

By

J.L. Frarey

And

R.F. Burchill

September 1976

U.S. ARMY MOBILITY EQUIPMENT
RESEARCH & DEVELOPMENT COMMAND
FORT BELVOIR, VIRGINIA 22060

Contract DAAG53-75-C-0287

SHAKER RESEARCH CORPORATION
NORTHWAY 10 EXECUTIVE PARK
BALLSTON LAKE, NEW YORK 12019

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SUMMARY

An earlier report prepared under Contract DAAK02-74-C-0084 investigated the feasibility of utilizing high frequency vibration analysis to diagnose power train condition in off-highway vehicles. The program was conducted on Caterpillar D8H crawler tractors and the results were promising. The present program investigates the commonality of the data from a D8 and a smaller crawler tractor the D7.

Based on the data from the D8, a spectral region between 20 KHz and 50 KHz was selected. This wide band width did not work for the D7 due to the presence of high level signals during steady-state operation in the 5 KHz to 35 KHz region. It was found that if the band width was limited to 40 KHz to 50 KHz, the diagnostic technique would apply equally well to both the D7 and D8 tractor.

An engineering model of a diagnostic system was built and tested that will operate on both the Caterpillar D7 and D8 crawler tractors.

PREFACE

This program was authorized under Contract DAAG53-75-C-0287 from the U.S. Army Mobility Equipment Research and Development Center. It is part of the overall effort by the U.S. Army to develop diagnostic tools for off-highway vehicles. The Army project engineer for this effort was Mr. James McLean AMXFB-HMM.

The authors wish to acknowledge the help of Mr. Milt Tatlock of Country Club Acres Inc. for the use of their D7 tractors for data gathering and system check-out and demonstration. Mr. Vince Bedell of the Southworth Machinery Company of Menands, New York also allowed us to use a D8H tractor for final checkout of the diagnostic system.

The Caterpillar Tractor Company of Peoria, Illinois also assisted us by furnishing the number of teeth on gears that were not given in the manuals.

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INTRODUCTION

Work was completed in early 1974 on an "Investigation of Off-road Vehicle Power Train Diagnostics Utilizing High Frequency Vibration Analysis", (Contract DAAK02-74-C-0084). The objective of this earlier work was to test a hypothesis that high frequency vibration analysis techniques were applicable to power train diagnostics. The result of this contract was that the technique showed promise. Signals obtained from a new tractor were those predicted for good components while those from older tractors contained characteristics that were predicted to be generated by defective components. All work done on this earlier contract involved the Caterpillar D-8 Crawler tractor.

While the results were promising, several questions remained. Some of these were:

1. Would the same type of data be generated by a different model tractor so that a diagnostic system could be applicable to a larger number of vehicles than just one model?
2. Could a simple system be designed and fabricated that would allow a diagnostic decision to be made immediately following the tractor test?
3. Most importantly, does the diagnostic technique really predict the mechanical condition of power train components?

The objective of the present contract work was to answer the first two questions. If in the limited sample of tractors analyzed, signals were present that indicated a seriously degraded power train component, the tractor could be torn down to get an indication as to the effectiveness of the diagnostic system. This was not the case and therefore question 3 remains unanswered.

It may seem that the priorities in answering the three questions are reversed; however, a fair test of a diagnostic technique can only be efficiently run if a prototype diagnostic system is available to allow analyses of a reasonable number of tractors in a short time.

In addition, the prediction of component condition must be based on the diagnostic hypothesis only and not inferred from the use of additional sophisticated data analysis techniques.

The objectives of this contract have been met. The single recommendation included in this report is to conduct a program to answer question number 3.

POWER TRAIN ANALYSIS

Description of Catepillar D7 and D8 Crawler Tractor Power Trains

The Caterpillar Crawler Tractor models that have been investigated in this and the previous contract include:

D-7E	All serial numbers
D-7F	S/N 5192 thru 5659
D-7F	S/N 5660 & up
D-7G	All S/N
D-8H	All S/N

All these tractors have the same schematic arrangement of the power drive components. Gearing ratios are different to account for different engine speeds - primary differences between four and six cylinder engines. The schematic of the Power Drive System is shown in Figure 1. The Torque Divider is directly coupled to the engine flywheel. For normal loads there is very little slip between input speed and output speed. However, the system has the capability of full slip (output shaft at rest) when the tractor is stalled. The power shift transmission is a planetary gearbox utilizing hydraulically actuated clutches. The output of the planetary portion of the power shift transmission is reduced by a spur gear reduction and the output shaft drives the cross shaft via a bevel gear. Following the steering clutch and brake, a double spur gear reduction drives the spocket.

The torque divider consists of a planetary gear set and hydraulic torque converter. The input drive is from the engine flywheel. Two paths are available for the transfer of torque through the system. The schematic of the system is shown in Figure 2. The engine flywheel drives the planetary sun gear and the impeller at the same speed. The impeller is fluid coupled to a turbine which in turn is mechanically coupled to the ring gear so that the ring gear rotates at the same speed as the turbine wheel. At no load, the turbine turns at the same speed as the impeller. Therefore, the planetary sun and ring gear are rotating at the same speed. The output is taken off the planet carrier and at no load the whole planetary system is rotating as a unit with no gear mesh action. In effect, the planetary set may be considered to

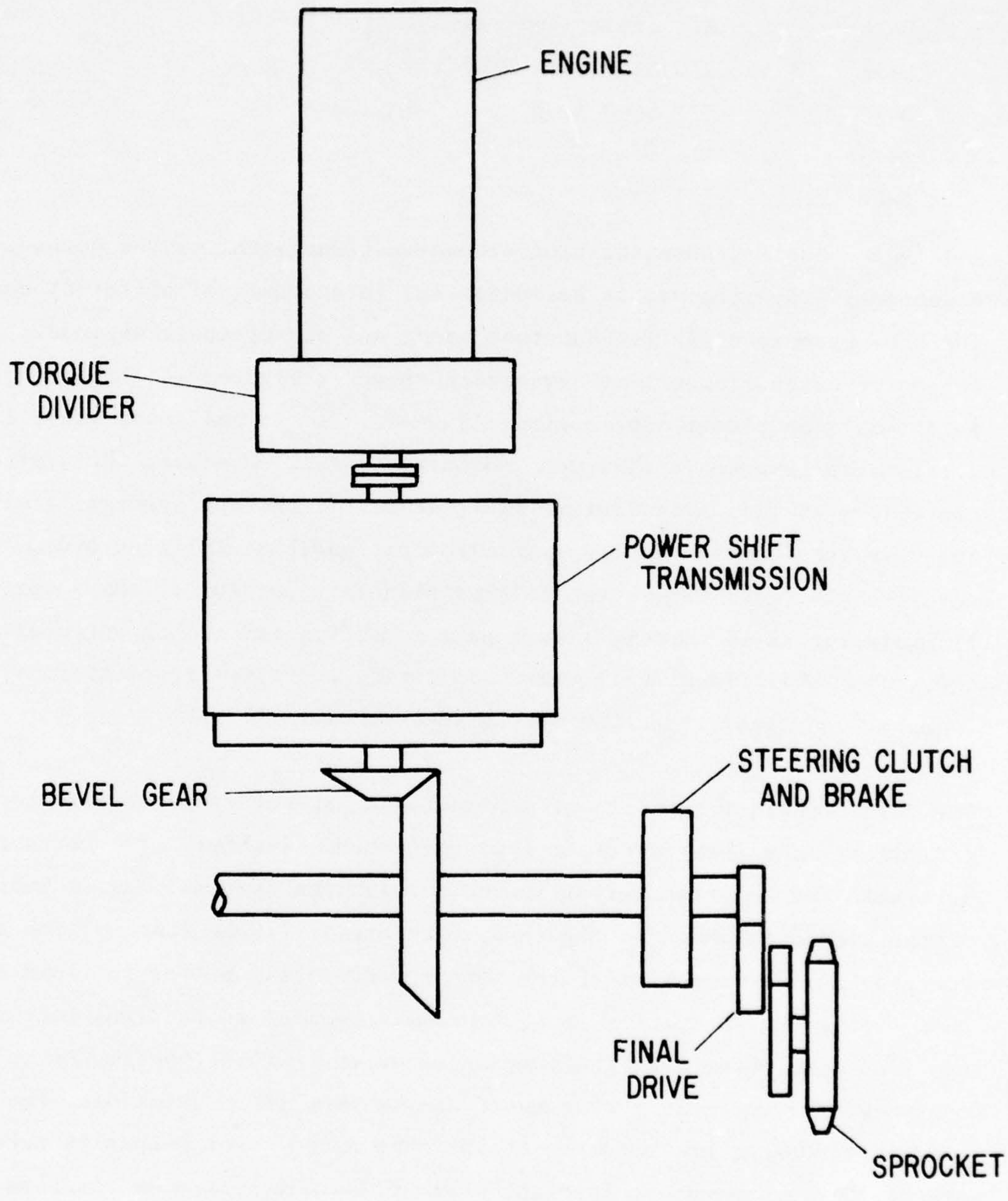


Figure 1 Power Train Schematic

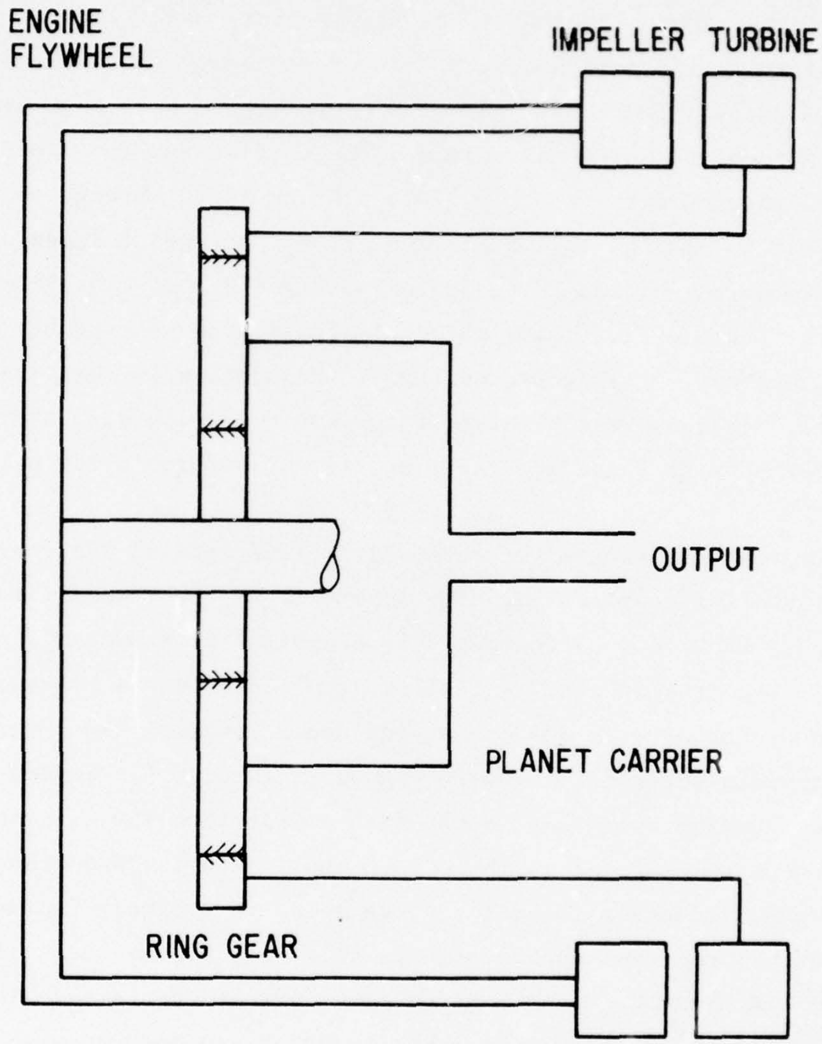


Figure 2 Torque Divider Schematic

be locked up. There is, therefore, no gear mesh signal generated at no load. As the output load is increased, the turbine and thus the ring gear slow down. The planet carrier, and therefore the output speed, slows down. As the turbine slows down with respec to the impeller, high torque is hydraulically transmitted to the turbine which tries to increase the turbine speed. In the limit of output power, the full stall condition, the output shaft, and thus the planet carrier does not turn and the turbine and ring gear will have reversed direction with very high torque transmitted through the ring gear to the output shaft. In the full stall condition, the engine speed drops from its normal 1380 rpm for the D8 to approximately 860 rpm. This design allows very little slippage at low loads while still allowing the tractor to be completely stalled without stalling the engine. In addition to shaft rotational frequencies, other sources of vibration signals are the planetary gear mesh signal (at other than no load) and the impeller and turbine blade passing frequencies.

The power shift transmission consists of five sets of planetary gear systems and a reduction gear system. The schematic of the transmission is shown in Figure 3. Each of the ring gears is supported by a hydraulically-actuated clutch. When the tractor is in gear, two clutches are engaged, either the forward or the reverse and one of the speed ranges. The power flow for the three forward gears are shown in Figures 4 through 6. Loaded gear sets are shaded. Reverse operation is the same except that the reverse planetary set is actuated which reverses the output shaft rotation and also produces a different gear reduction so that reverse speed is slightly faster for each gear selected. When a new speed range is selected, the previous clutch is disengaged and the new clutch engaged in very rapid sequence. This forces the transmission to try to almost instanteaneously change speed. While the torque divider may slip to reduce the applied loads at the gears and bearings, the shift transient nevertheless imposes very high loads on the system for a short time duration. The basis of the diagnostic system is that for normal power train components, this high shock load will cause the system to ring at a high frequency for a very short time duration. For abnormal components, the excitation of this high frequency resonance will continue for a longer period and high level signals will persist for steady-state operation. A further

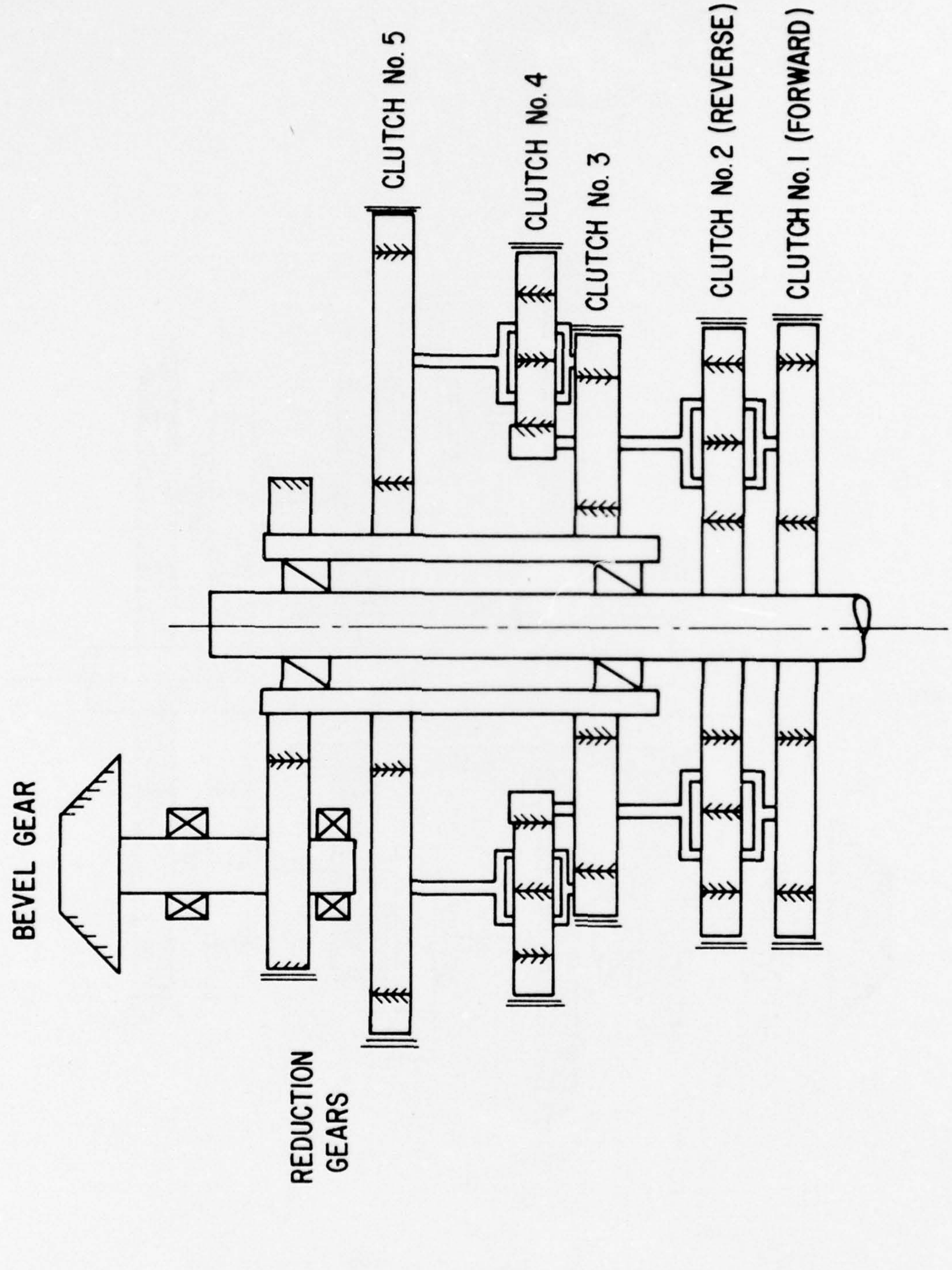


Figure 3 Power Shift Transmission Schematic

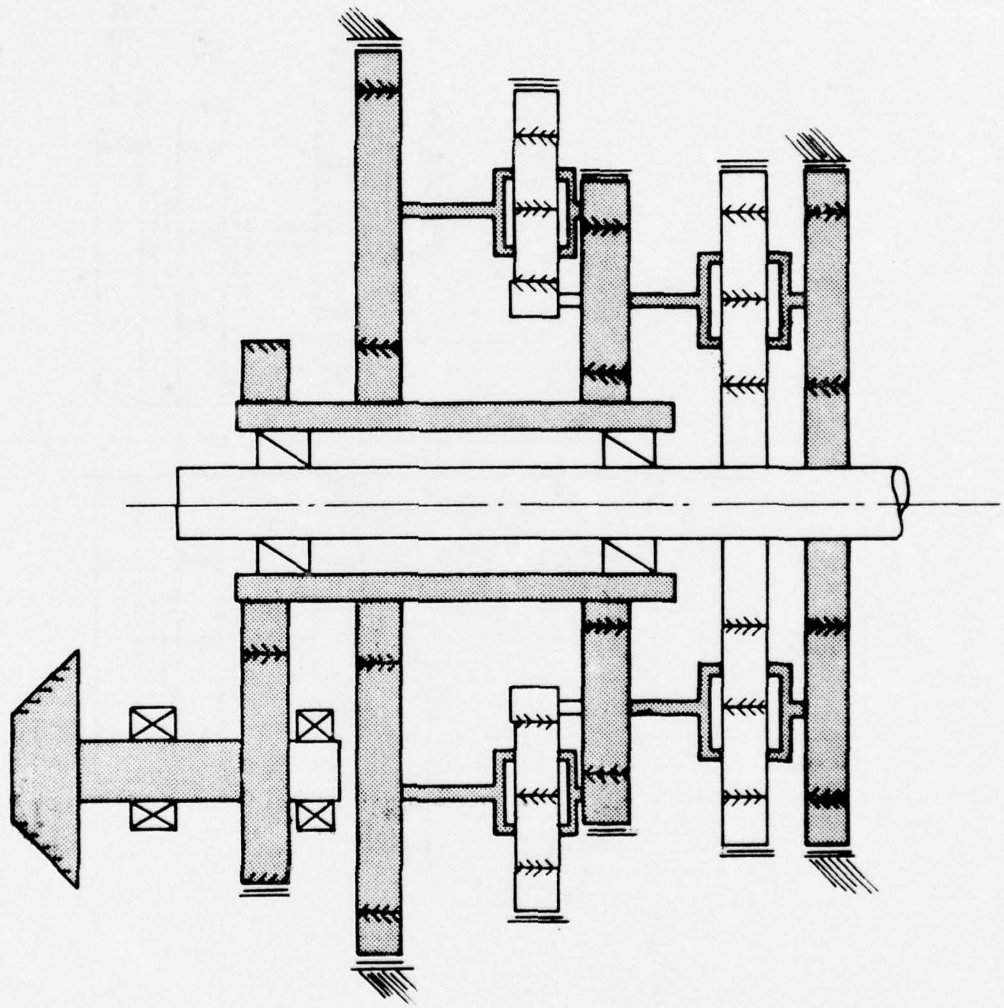


Figure 4 Power Flow 1st Forward Gear

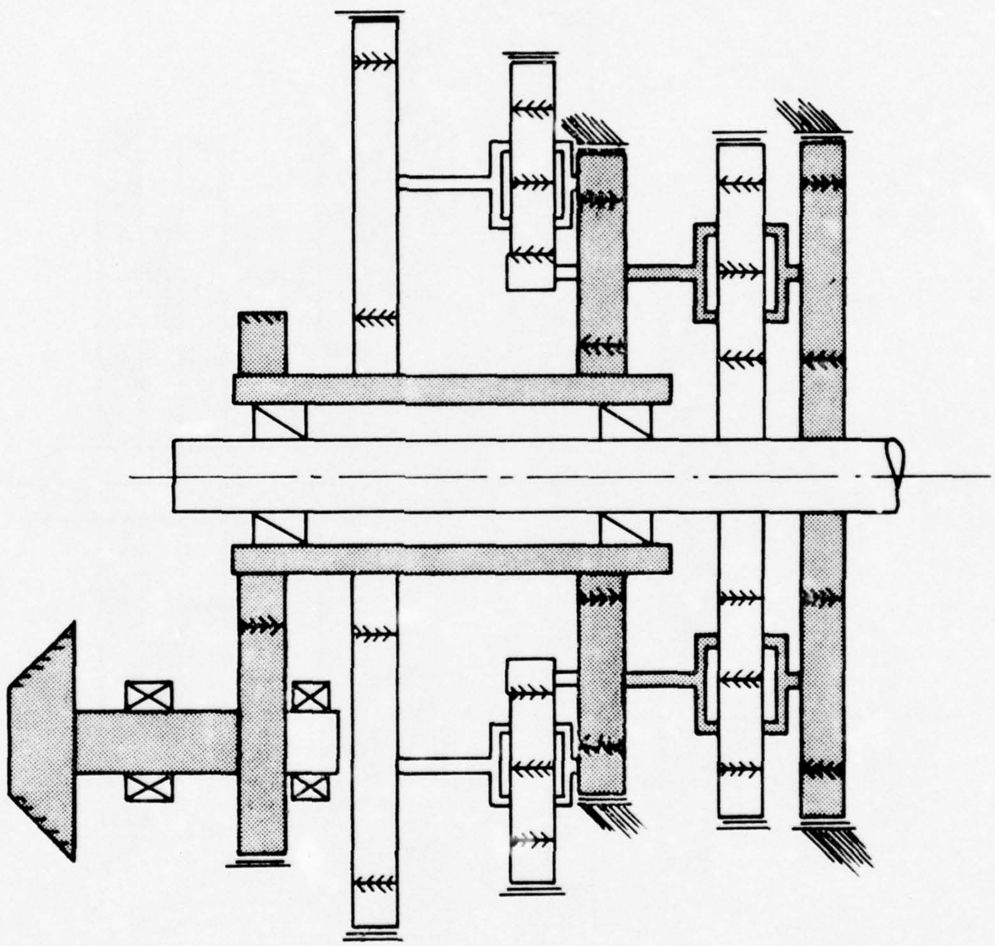


Figure 5 Power Flow 2nd Forward Gear

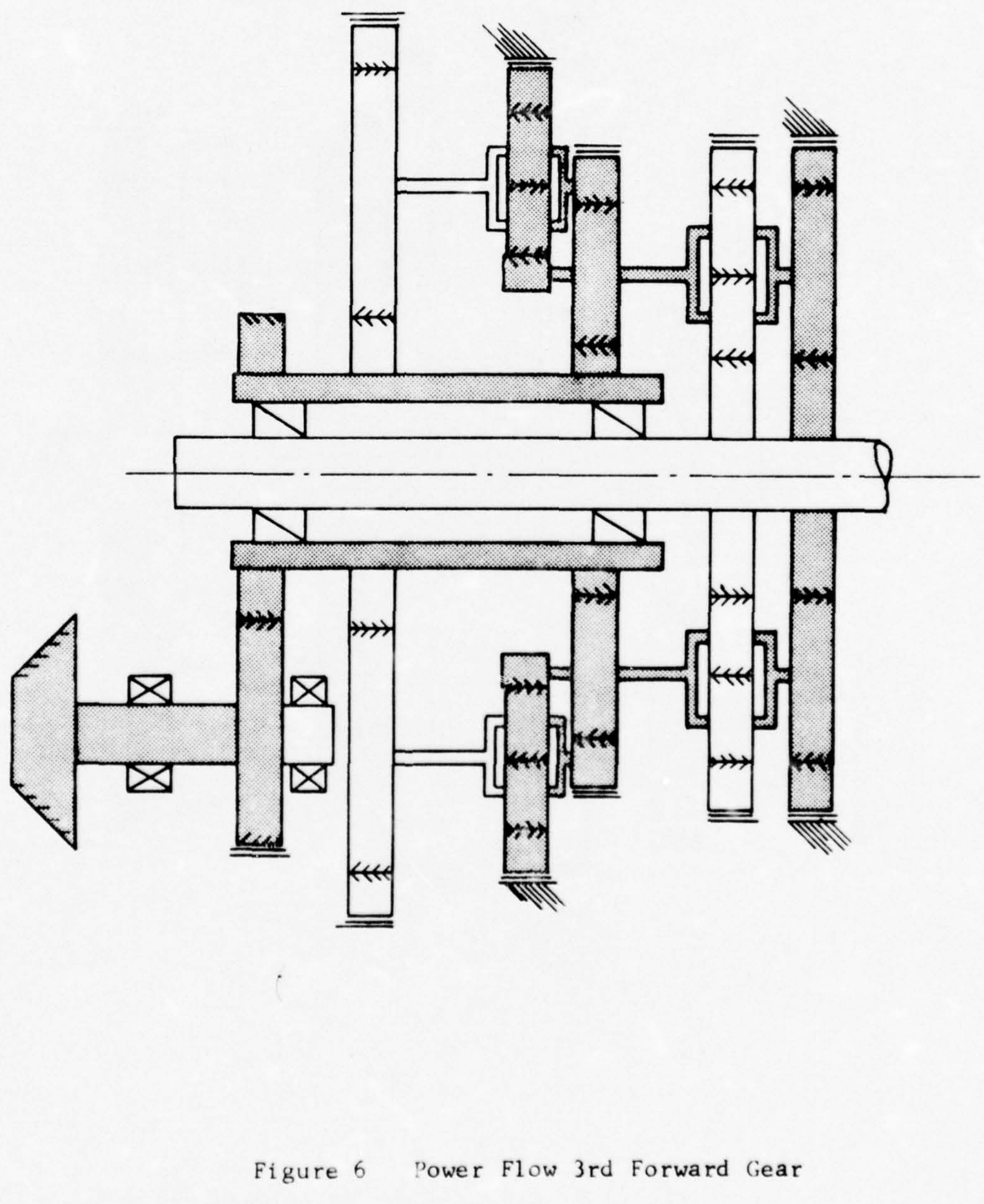


Figure 6 Power Flow 3rd Forward Gear

analysis may be made in the lab by examining the frequency content of the demodulated high frequency resonance. Gear mesh frequencies may be calculated and compared to the signals present in the demodulated spectra. In this way, identification of defective components may be made.

Calculation of Gear Mesh Frequencies

Power train ratios, gear mesh frequencies and the rotational speeds of planet gears and planet carriers were calculated for each tractor model. These calculations were made from information in the Parts and Service Manual and additional information furnished by Caterpillar. Tables 1 through 5 contain these calculations for each of the models and variations of models. The top set of numbers are the ratios and cumulative ratios at various points in the power train. The front ratio is the ratio of either the forward or reverse planetary. The rear ratio is the ratio of the last three planetaries or rear carrier. Total planet is the ratio through both carriers. Transmission out is the cumulative ratio between the transmission input shaft and the transmission output shaft after the spur gear stage. The axle data is the cumulative ratio for the axle after the bevel stage while the spocket drive is again the cumulative ratio at the spocket. A positive sign for the numbers on all tables means that rotation is in the same direction as the transmission input shaft.

The next line in the table is the engine RPM used for all the following frequency calculations. The loaded planetary frequencies are zero if the planet does not carry any load. These unloaded planetaries are in mesh and turning and the frequencies could be calculated.

Following the two planet carriers in the transmission, there are four additional gear meshes in the power train. These gear mesh frequencies are the next set of numbers. Next, individual planet gear rotational frequencies are given. For the second and fourth planetary, A & B indicate that there are two planets in series in this stage. Finally the front and back carrier rotational speeds are given for the transmission.

TRACTØR MØDEL D8H

PØWER TRAIN RATIØS

SPEED	FRØNT RATIØ	REAR RATIØ	TØTAL PLANET	XMSN ØUT	AXLE RATIØ	SPRØCKET DRIVE
1F	+0.3623	+2.2535	+0.8164	+0.5715	+.2449	+.01425
2F	+0.3623	+4.0000	+1.4492	+1.0144	+.4347	+.02529
3F	+0.3623	+6.2941	+2.2804	+1.5963	+.6841	+.03980
1R	-0.4500	+2.2535	-1.0140	-0.7098	-.3042	-.01770
2R	-0.4500	+4.0000	-1.8000	-1.2600	-.5400	-.03141
3R	-0.4500	+6.2941	-2.8323	-1.9826	-.8497	-.04943

RPM = 1280

GEAR MESH FREQUENCIES--HZ--LØADED PLANETARIES

SPEED	1 PLANET	2 PLANET	3 PLANET	4 PLANET	5 PLANET
1F	680	0	261	0	400
2F	680	0	626	0	0
3F	680	0	1104	531	0
1R	0	835	324	0	497
2R	0	835	777	0	0
3R	0	835	1372	660	0

GEAR MESH FREQUENCIES--HZ--SPURS AND BEVEL

SPEED	XMISN ØUT	BEVEL	TØP F.D.	BØTTØM F.D.
1F	609	256	62.7	16.7
2F	1082	454	111.3	29.6
3F	1702	715	175.1	46.7
1R	757	318	77.8	20.7
2R	1344	564	138.2	36.8
3R	2114	888	217.5	58.0

PLANET RØTATIØNAL FREQUENCY --RPS--

SPEED	1	2A	2B	3	4A	4B	5
1F	-35.79	-13.60	+19.33	-9.68	-5.99	+6.62	-13.80
2F	-35.79	-13.60	+19.33	-23.18	-14.35	+15.86	-33.05
3F	-35.79	-13.60	+19.33	-40.92	-25.33	+27.99	-58.32
1R	-81.40	-30.93	+43.95	+12.03	+7.44	-8.23	+17.15
2R	-81.40	-30.93	+43.95	+28.80	+17.82	-19.70	+41.04
3R	-81.40	-30.93	+43.95	+50.82	+31.46	-34.77	+72.43

CARRIER RØTATIØNAL FREQUENCIES--RPS--

SPEED	FRØNT CARRIER	BACK CARRIER
1F	+7.729	+4.499
2F	+7.729	+0.000
3F	+7.729	-5.910
1R	-9.599	-5.588
2R	-9.599	+0.000
3R	-9.599	+7.341

TRACTØR MØDEL D7G

PØWER TRAI RATIØS

SPEED	FRØNT RATIØ	REAR RATIØ	TØTAL PLANET	XMSN ØUT	AXLE RATIØ	SPRØCKET DRIVE
1F	+0.3333	+1.9932	+0.6644	+0.4932	+0.1517	+.00894
2F	+0.3333	+3.9600	+1.3200	+0.9799	+0.3015	+.01776
3F	+0.3333	+6.2622	+2.0874	+1.5497	+0.4768	+.02809
1R	-0.4035	+1.9932	-0.8043	-0.5971	-0.1837	-.01082
2R	-0.4035	+3.9600	-1.5978	-1.1363	-0.3650	-.02150
3R	-0.4035	+6.2622	-2.5268	-1.8760	-0.5772	-.03401

RPM = 2000

GEAR MESH FREQUENCIES--HZ--LØADED PLANETARIES

SPEED	1 PLANET	2 PLANET	3 PLANET	4 PLANET	5 PLANET
1F	888	0	275	0	590
2F	888	0	322	0	0
3F	888	0	1461	691	0
1R	0	1076	334	0	714
2R	0	1076	995	0	0
3R	0	1076	1769	836	0

GEAR MESH FREQUENCIES--HZ--SPURS AND BEVEL

SPEED	XMISN ØUT	BEVEL	TØP F.D.	BØTTØM F.D.
1F	1085	263	60.7	15.5
2F	2156	522	120.6	30.7
3F	3409	826	190.7	48.6
1R	1313	318	73.4	18.7
2R	2609	632	146.0	37.2
3R	4127	1000	230.8	58.9

PLANET RØTATIØNAL FREQUENCY --RPS--

SPEED	1	2A	2B	3	4A	4B	5
1F	-44.44	-19.65	+25.55	-11.49	-6.52	+6.52	-29.53
2F	-44.44	-19.65	+25.55	-34.25	-19.44	+19.44	-87.99
3F	-44.44	-19.65	+25.55	-60.90	-34.56	+34.56	-156.44
1R	-93.56	-41.38	+53.80	+13.91	+7.89	-7.89	+35.74
2R	-93.56	-41.38	+53.80	+41.47	+23.53	-23.53	+106.52
3R	-93.56	-41.38	+53.80	+73.72	+41.84	-41.84	+189.38

CARRIER RØTATIØNAL FREQUENCIES--RPS--

SPEED	FRØNT CARRIER	BACK CARRIER
1F	+11.111	+7.382
2F	+11.111	+0.000
3F	+11.111	-8.641
1R	-13.450	-8.936
2R	-13.450	+0.000
3R	-13.450	+10.461

TABLE 2

TRACTØR MØDEL D7F 5660 & UP

POWER TRA. RATIØS

SPEED	FRØNT RATIØ	REAR RATIØ	TØTAL PLANET	XMSN ØUT	AXLE RATIØ	SPRØCKET DRIVE
1F	+0.3333	+1.9932	+0.6644	+0.4592	+0.1413	+.00832
2F	+0.3333	+3.9600	+1.3200	+0.9123	+0.2807	+.01654
3F	+0.3333	+6.2622	+2.0874	+1.4427	+0.4439	+.02615
1R	-0.4035	+1.9932	-0.8043	-0.5559	-0.1710	-.01007
2R	-0.4035	+3.9600	-1.5978	-1.1044	-0.3398	-.02002
3R	-0.4035	+6.2622	-2.5268	-1.7465	-0.5373	-.03166

RPM = 2000

GEAR MESH FREQUENCIES--HZ--LØADED PLANETARIES

SPEED	1 PLANET	2 PLANET	3 PLANET	4 PLANET	5 PLANET
1F	888	0	275	0	590
2F	888	0	822	0	0
3F	888	0	1461	691	0
1R	0	1076	334	0	714
2R	0	1076	995	0	0
3R	0	1076	1769	836	0

GEAR MESH FREQUENCIES--HZ--SPURS AND BEVEL

SPEED	XMISN ØUT	BEVEL	TØP F.D.	BØTTØM F.D.
1F	1040	244	56.5	14.4
2F	2068	486	112.2	28.6
3F	3270	769	177.5	45.3
1R	1260	296	68.4	17.4
2R	2503	589	135.9	34.7
3R	3958	931	214.9	54.8

PLANET RØTATIØNAL FREQUENCY --RPS--

SPEED	1	2A	2B	3	4A	4B	5
1F	-44.44	-19.65	+25.55	-11.49	-6.52	+6.52	-29.53
2F	-44.44	-19.65	+25.55	-34.25	-19.44	+19.44	-87.99
3F	-44.44	-19.65	+25.55	-60.90	-34.56	+34.56	-156.44
1R	-93.56	-41.38	+53.80	+13.91	+7.39	-7.89	+35.74
2R	-93.56	-41.38	+53.80	+41.47	+23.53	-23.53	+106.52
3R	-93.56	-41.38	+53.80	+73.72	+41.84	-41.84	+189.38

CARRIER RØTATIØNAL FREQUENCIES--RPS--

SPEED	FRØNT CARRIER	BACK CARRIER
1F	+11.111	+7.382
2F	+11.111	+0.000
3F	+11.111	-8.641
1R	-13.450	-8.936
2R	-13.450	+0.000
3R	-13.450	+10.461

TABLE 3

TRACTØR MØDEL D7F 5192 - 5659

PØWER TRAIN RATIØS

SPEED	FRØNT RATIØ	REAR RATIØ	TØTAL PLANET	XMSN ØUT	AXLE RATIØ	SPRØCKET DRIVE
1F	+0.3605	+2.2165	+0.7991	+0.5203	+0.1592	+.00938
2F	+0.3605	+3.9000	+1.4061	+0.9156	+0.2802	+.01651
3F	+0.3605	+6.3450	+2.2876	+1.4896	+0.4560	+.02686
1R	-0.4242	+2.2165	-0.9403	-0.6123	-0.1874	-.01104
2R	-0.4242	+3.9000	-1.6545	-1.0773	-0.3298	-.01943
3R	-0.4242	+6.3450	-2.6918	-1.7528	-0.5365	-.03161

RPM = 2000

GEAR MESH FREQUENCIES--HZ--LØADED PLANETARIES

SPEED	1 PLANET	2 PLANET	3 PLANET	4 PLANET	5 PLANET
1F	1129	0	438	0	648
2F	1129	0	1045	0	0
3F	1129	0	1927	952	0
1R	0	1329	516	0	763
2R	0	1329	1230	0	0
3R	0	1329	2267	1120	0

GEAR MESH FREQUENCIES--HZ--SPURS AND BEVEL

SPEED	XMISN ØUT	BEVEL	TØP F.D.	BØTTØM F.D.
1F	745	260	63.7	16.2
2F	1312	457	112.1	28.6
3F	2135	744	182.4	46.5
1R	877	306	74.9	19.1
2R	1544	536	131.9	33.6
3R	2512	876	214.6	54.8

PLANET RØTATIØNAL FREQUENCY --RPS--

SPEED	1	2A	2B	3	4A	4B	5
1F	-56.48	-19.89	+29.84	-15.66	-10.83	+10.83	-21.62
2F	-56.48	-19.89	+29.84	-37.34	-25.83	+25.83	-51.55
3F	-56.48	-19.89	+29.84	-68.82	-47.62	+47.62	-95.02
1R	-125.80	-44.30	+66.46	+18.43	+12.75	-12.75	+25.44
2R	-125.80	-44.30	+66.46	+43.93	+30.40	-30.40	+60.66
3R	-125.80	-44.30	+66.46	+80.98	+56.03	-56.03	+111.81

CARRIER RØTATIØNAL FREQUENCIES--RPS--

SPEED	FRØNT CARRIER	BACK CARRIER
1F	+12.018	+6.976
2F	+12.018	+0.000
3F	+12.018	-10.132
1R	-14.141	-8.209
2R	-14.141	+0.000
3R	-14.141	+11.923

TABLE 4

PØWER TRAIN RATIØS

SPEED	FRØNT RATIØ	REAR RATIØ	TØTAL PLANET	XMSN ØUT	AXLE RATIØ	SPRØCKET DRIVE
1F	+0.3605	+2.2165	+0.7991	+0.8931	+0.7051	+.04154
2F	+0.3605	+3.9000	+1.4061	+1.5715	+1.2406	+.07310
3F	+0.3605	+6.3450	+2.2876	+2.5568	+2.0185	+.11893
1R	-0.4242	+2.2165	-0.9403	-1.0509	-0.8297	-.04888
2R	-0.4242	+3.9000	-1.6545	-1.8491	-1.4598	-.08601
3R	-0.4242	+6.3450	-2.6918	-3.0085	-2.3751	-.13994

RPM = 1200

GEAR MESH FREQUENCIES--HZ--LØADED PLANETARIES

SPEED	1 PLANET	2 PLANET	3 PLANET	4 PLANET	5 PLANET
1F	677	0	263	0	389
2F	677	0	627	0	0
3F	677	0	1156	571	0
1R	0	797	309	0	458
2R	0	797	738	0	0
3R	0	797	1360	672	0

GEAR MESH FREQUENCIES--HZ--SPURS AND BEVEL

SPEED	XMISN ØUT	BEVEL	TØP F.D.	BØTTØM F.D.
1F	607	267	169.2	43.2
2F	1068	471	297.7	76.0
3F	1738	767	484.4	123.6
1R	714	315	199.1	50.8
2R	1257	554	350.3	89.4
3R	2045	902	570.0	145.5

PLANET RØTATIØNAL FREQUENCY --RPS--

SPEED	1	2A	2B	3	4A	4B	5
1F	-33.89	-11.93	+17.90	-9.39	-6.50	+6.50	-12.97
2F	-33.89	-11.93	+17.90	-22.40	-15.50	+15.50	-30.93
3F	-33.89	-11.93	+17.90	-41.29	-28.57	+28.57	-57.01
1R	-75.48	-26.58	+39.87	+11.05	+7.65	-7.65	+15.26
2R	-75.48	-26.58	+39.87	+26.36	+18.24	-18.24	+36.40
3R	-75.48	-26.58	+39.87	+48.59	+33.62	-33.62	+67.09

CARRIER RØTATIØNAL FREQUENCIES--RPS--

SPEED	FRØNT CARRIER	BACK CARRIER
1F	+7.210	+4.186
2F	+7.210	+0.000
3F	+7.210	-6.079
1R	-8.484	-4.925
2R	-8.484	+0.000
3R	-8.484	+7.153

TABLE 5

DATA ANALYSIS

The objective of the data analysis frequencies for the D7 series of tractors was to determine the common elements between the D8 tractor and the D7 tractor. The D8 tractor was investigated in a previous contract (Reference 1). The instrumentation system used in this program is essentially the same as was used in the previous program and will be treated only briefly.

The recording instrumentation is shown in Figure 7. The accelerometers have a mounted resonance above 70 KHz and the voltage follower and amplifier have response in excess of 100 KHz. The portable scope was used to monitor input levels at the input to the tape recorder. The tape recorder was run at 15 ips which produced a direct response from 100 Hz to 50 KHz. The complete system was powered off the tractor battery using a 12 VDC to 120 V 60 Hz inverter. Accelerometer locations were changed to various locations to investigate the relative signal characteristics as well as to compare D7 data with earlier D8 data.

The data reduction instrumentation was again similar to that used in the previous work and is shown in Figure 8. The top arrangement is used to investigate the overall spectra content of the accelerometer signal for steady-state operation and during the shift transients. The bottom arrangement is used to plot the overall signal amplitude during a shift transient.

Figure 9 and 10 are taken from the earlier work on the D8 tractor showing steady state and shift transient data typical of the D8 tractor. Steady state operation shows very low amplitude in the spectral region above 15 KHz.
The shift transient occurs in the 40 KHz to 60 KHz region of the spectra. If the data were filtered above 20 KHz, and the amplitude vs time traces were to be plotted, a transient would be seen for each shift. The mounted accelerometer resonance however can occur in the region from 50 KHz to 100 KHz depending on the quality of the bond between the accelerometer and the mounting location on the tractor. The presence of the transient response of the accelerometer at

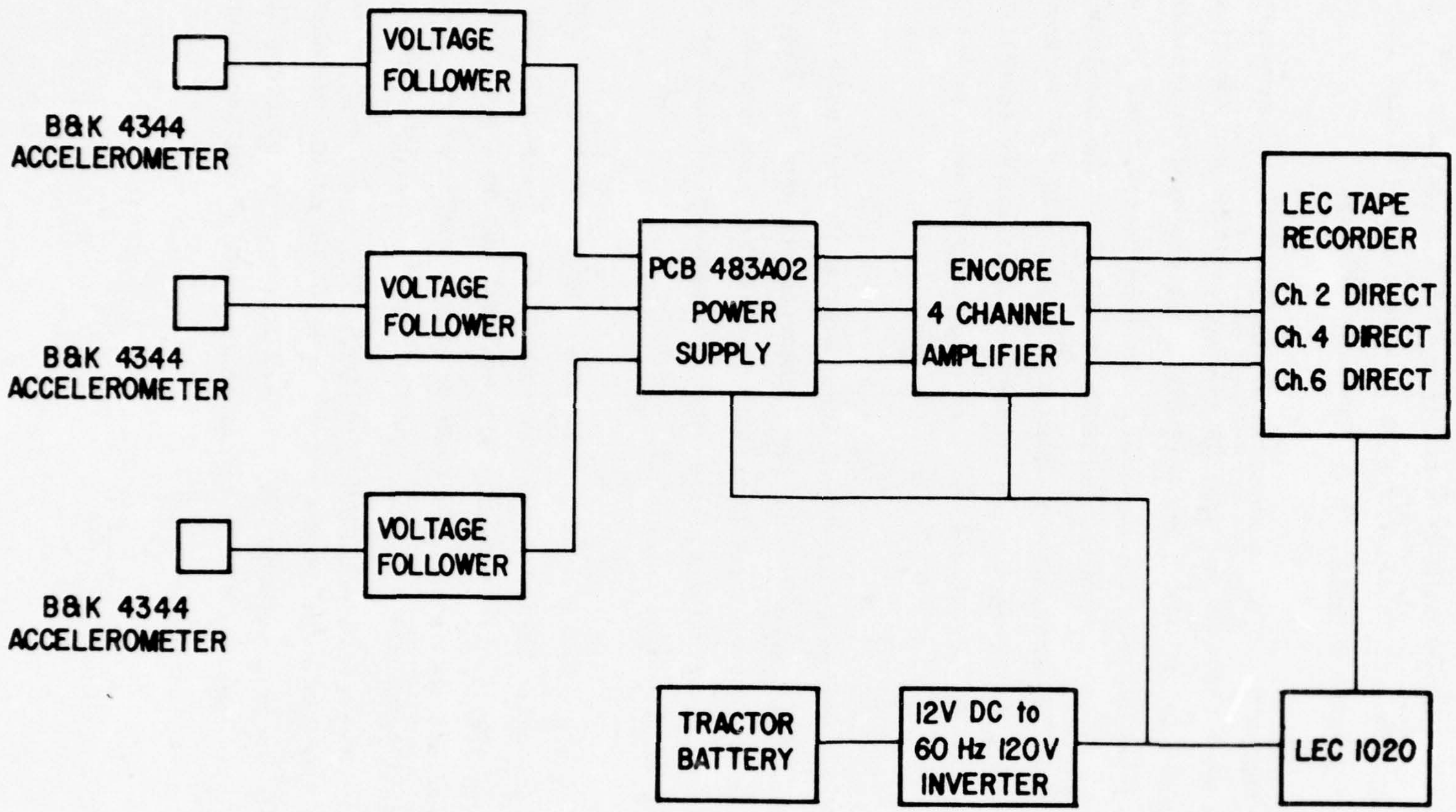
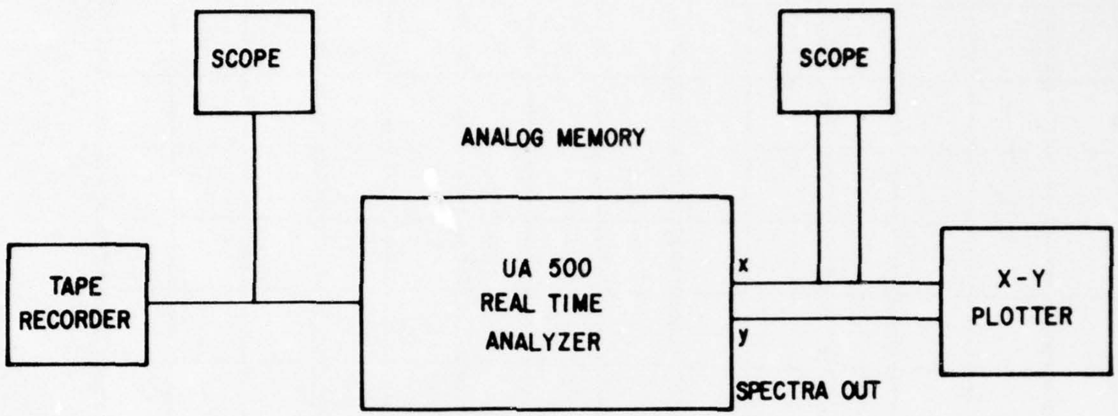
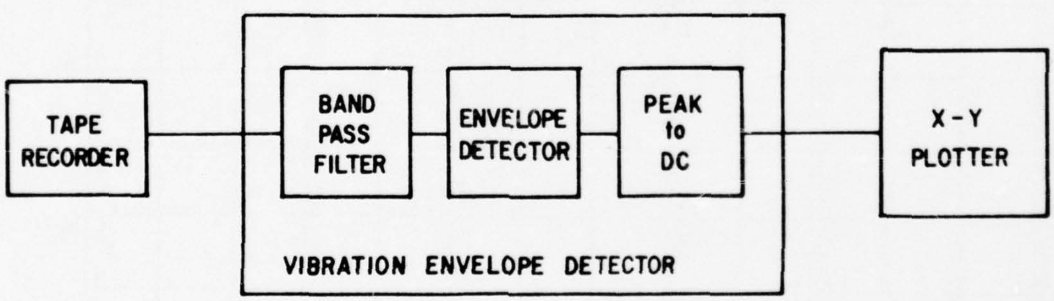


Figure 7 Recording Instrumentation Configuration



(a) HIGH-FREQUENCY SPECTRAL PLOTS



(b) AMPLITUDE VERSUS TIME PLOTS

Figure 8 Data Reduction Instrumentation

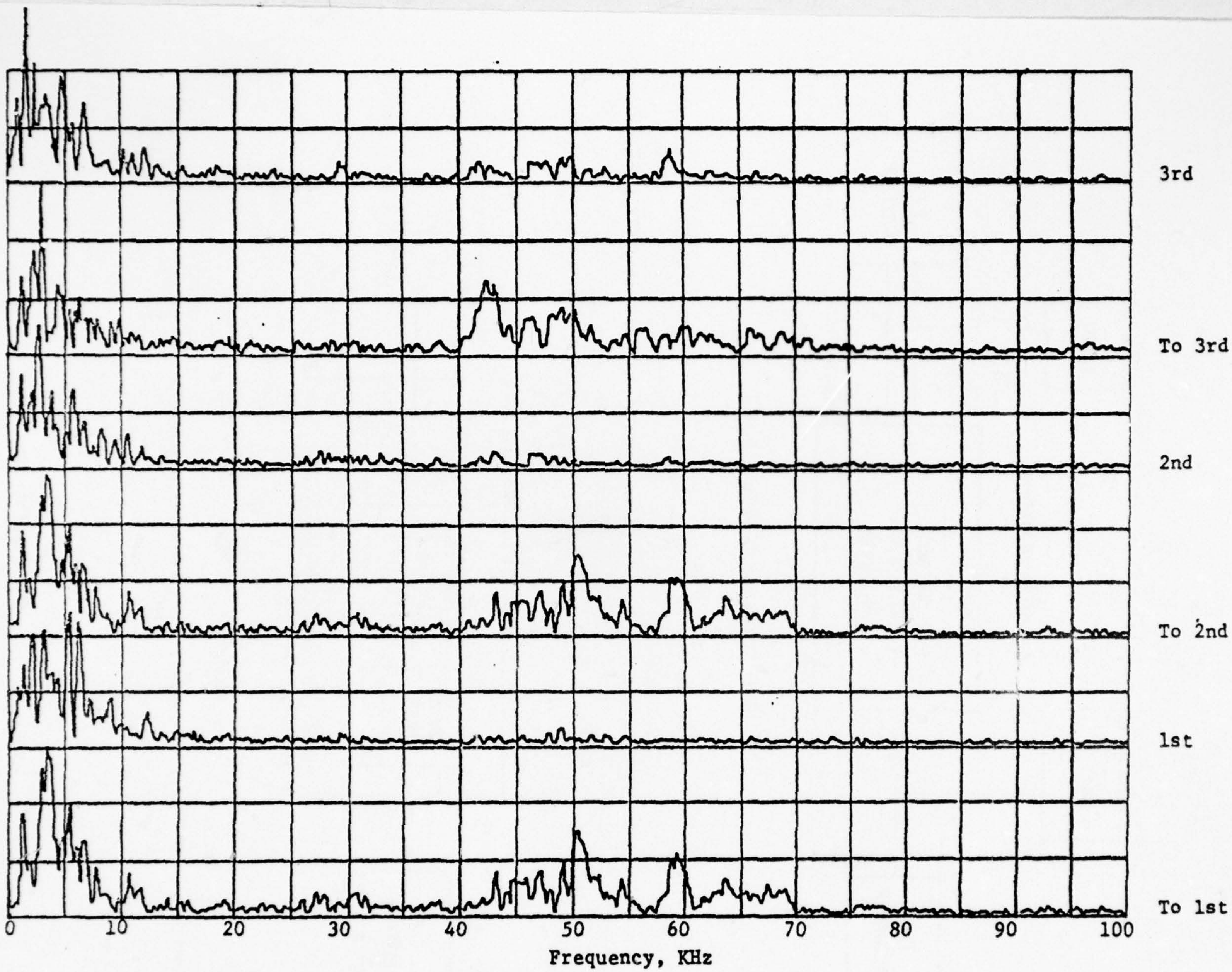


Figure 9 Transmission Forward Shift Transient D8H

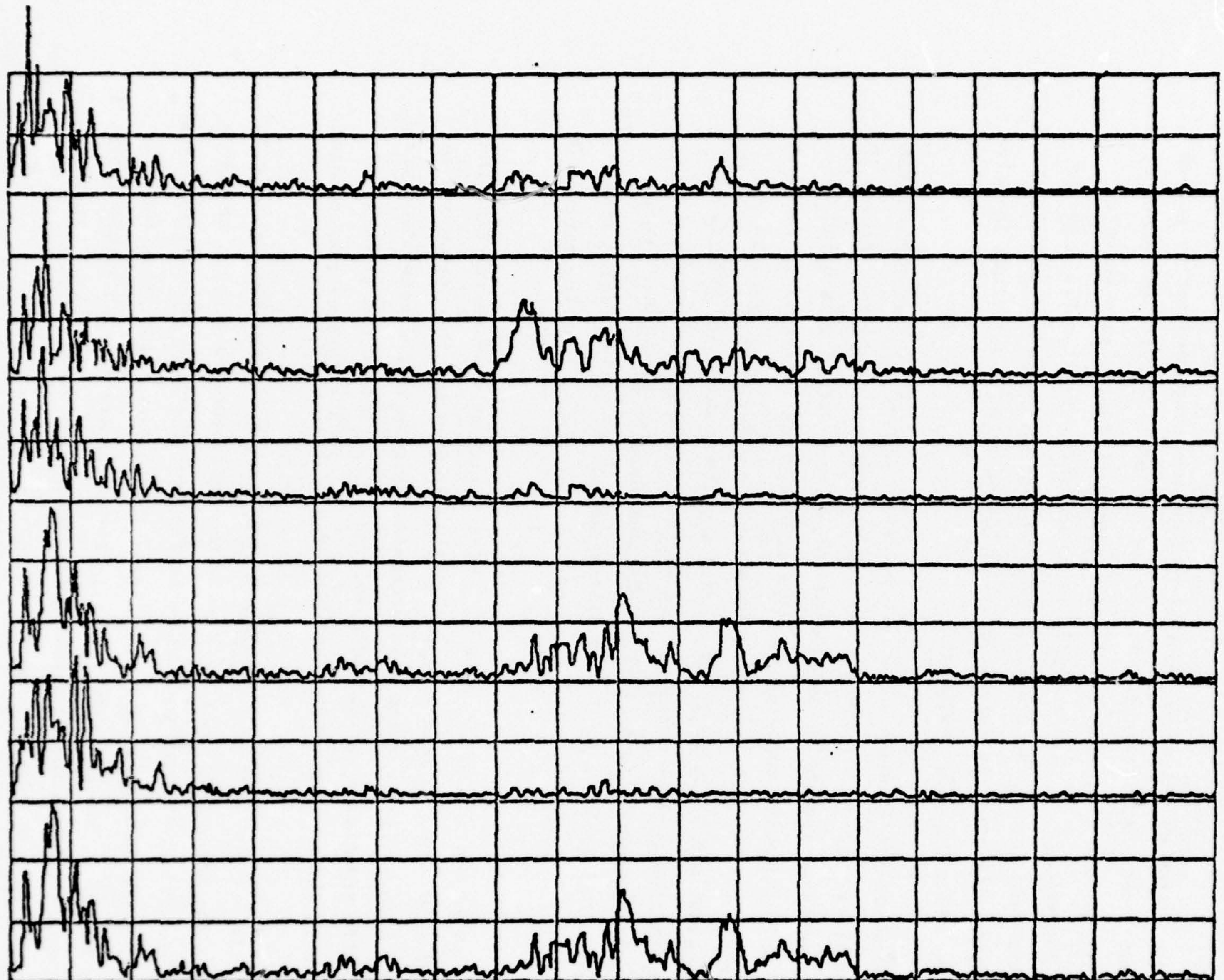


Figure 10 Transmission Reverse Shift Transient D8H

its resonance could affect the data greatly if only the amplitude vs time trace were observed without the spectral data back up. For this reason, D8 data was filtered between 20 KHz and 50 KHz.

The recording system as set up for the D7 tractor was therefore set to record to 50 KHz and reject data above 50 KHz. After two D7 tractors were recorded, the data was analyzed using the bottom arrangement of instruments shown in Figure 8. The results of this analysis is shown in Figure 11. Based on the criteria developed for the D8 tractor, one would be forced to conclude that the tractor was defective in third gear. (Note the rise in amplitude in third gear forward and third gear reverse.)

In order to check the location of the resonance, the data was run through the spectrum analyzer for steady-state operation and shift transients. Figure 12 shows this analysis for the D7 and should be compared with Figure 9 and 10. One thing is immediately apparent when the steady-state vibration is analyzed in third gear; that is, the extremely high level signals that occur in the spectral region from 5 KHz to 35 KHz. This amplitude increases with speed after the transient. If the region from 40 KH to 50 KHz is examined, it is seen that this spectral region follows the same pattern as the D8. It was therefore decided to use a band pass filter restricted to the region from 40 KHz to 50 KHz rather than 20 KHz to 50 KHz as appeared from the earlier D8 data.

Figure 13 shows the time history of the shift transient when restricting the data to 40 - 50 KHz. D8 data from the earlier program was then rerun and no difference from earlier results were obtained. Figure 14 compares the transmission shift transients for two D8's and two D7's. The bottom trace is for the D8 run in the earlier program that had a suspected defective transmission.

Figure 15 is a comparison of data taken on the torque divider for two D7's and two D8's. The background data for the D7 is consistently higher in amplitude than for the D8's. This is due mainly to the fact that the four cylinder engine of the D7E produces higher shock loading of the torque divider than the six cylinder engine of the D8. The top trace shows reasonably high levels in third gear. This is not considered significant because the operation of the torque divider is identical for each speed range.

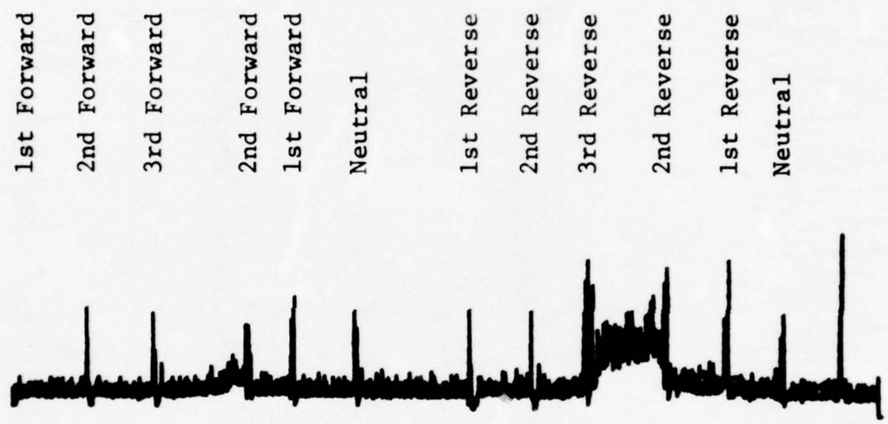


Figure 11 Transmission Trace D7E

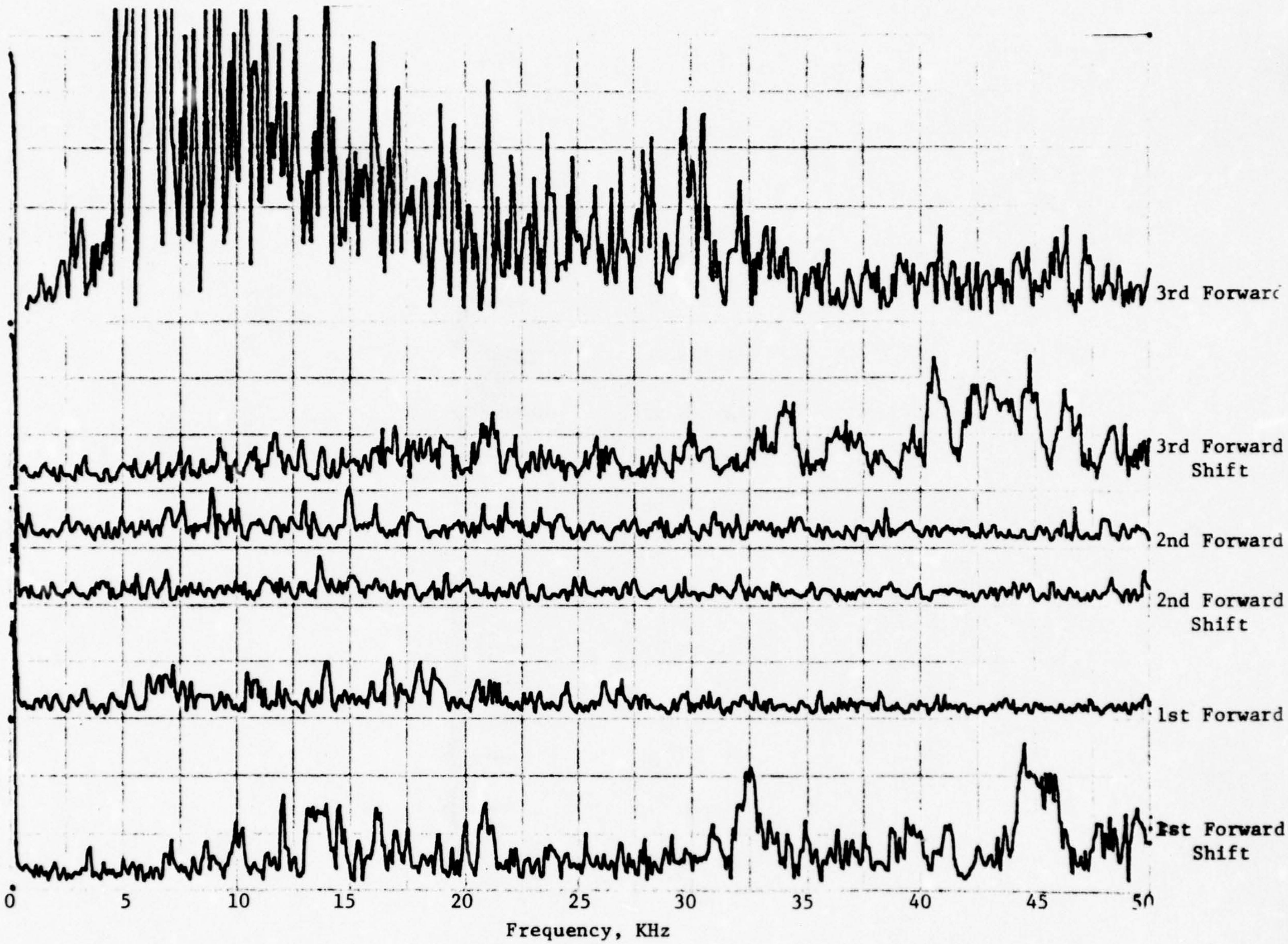


Figure 12 Transmission Forward Shift Transient D7E

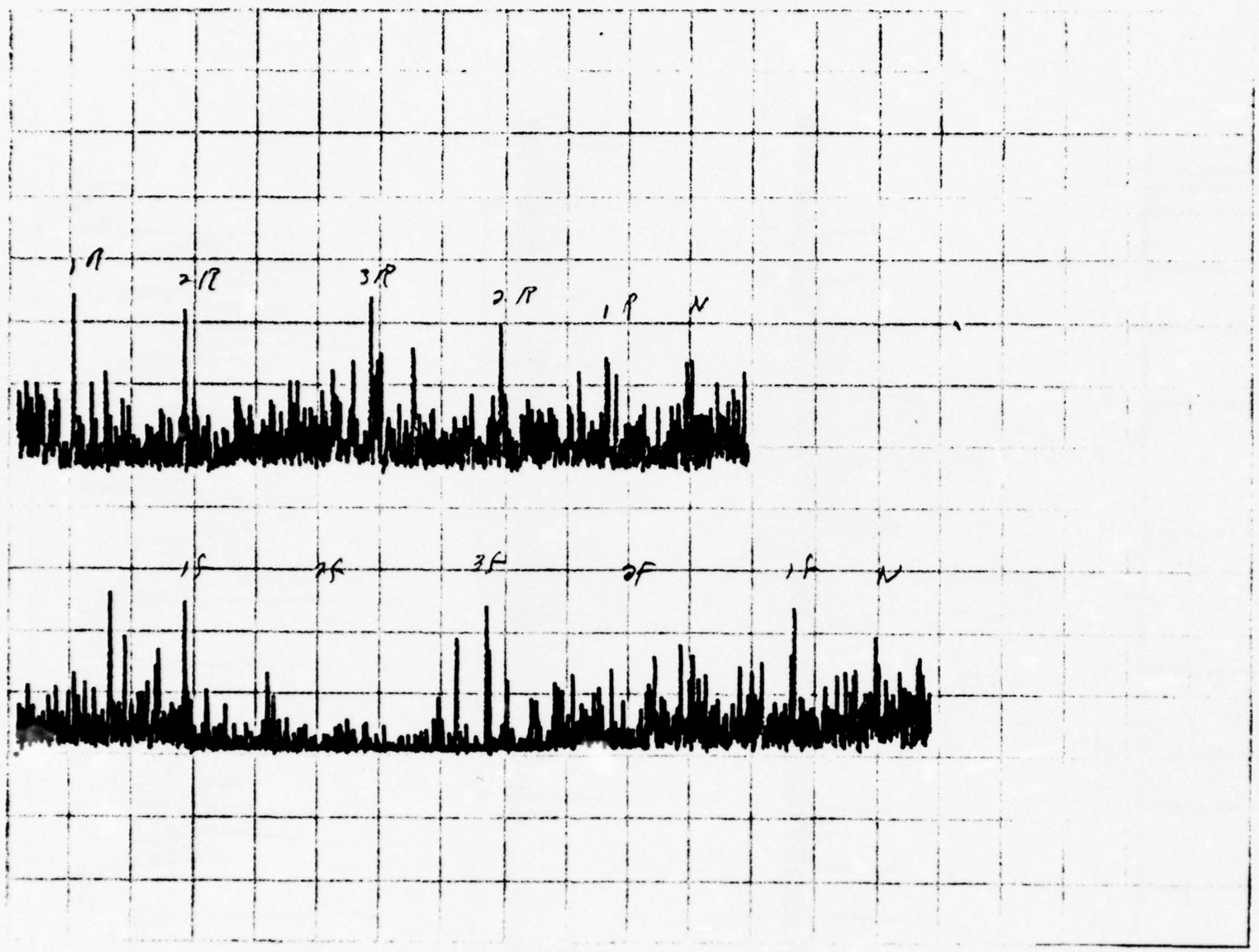


Figure 13 Transmission Trace (40 KHz - 50 KHz) D7E

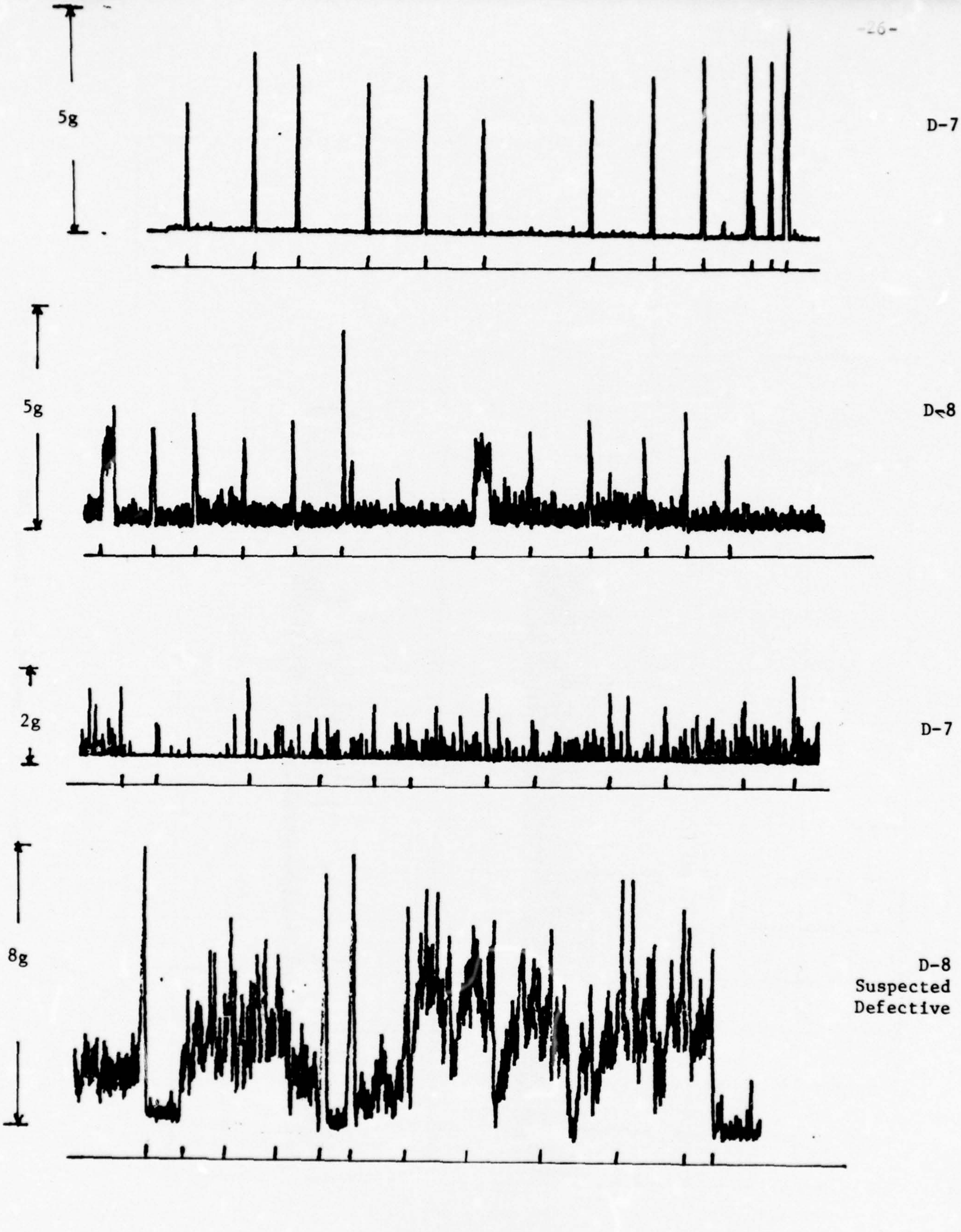


Figure 14 Transmission Trace D7E and D8H

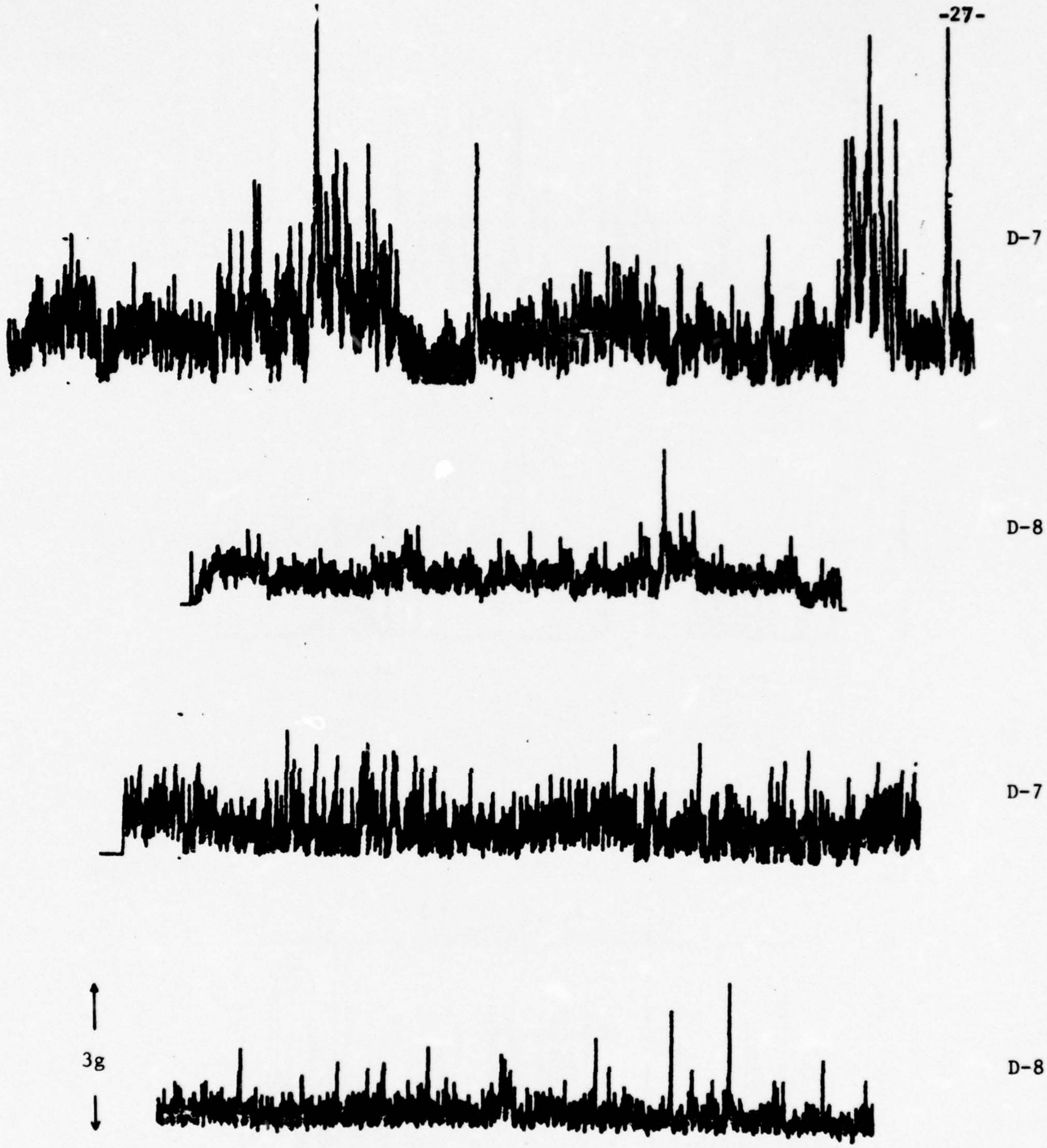


Figure 15 Torque Divider Trace D7E and D8H

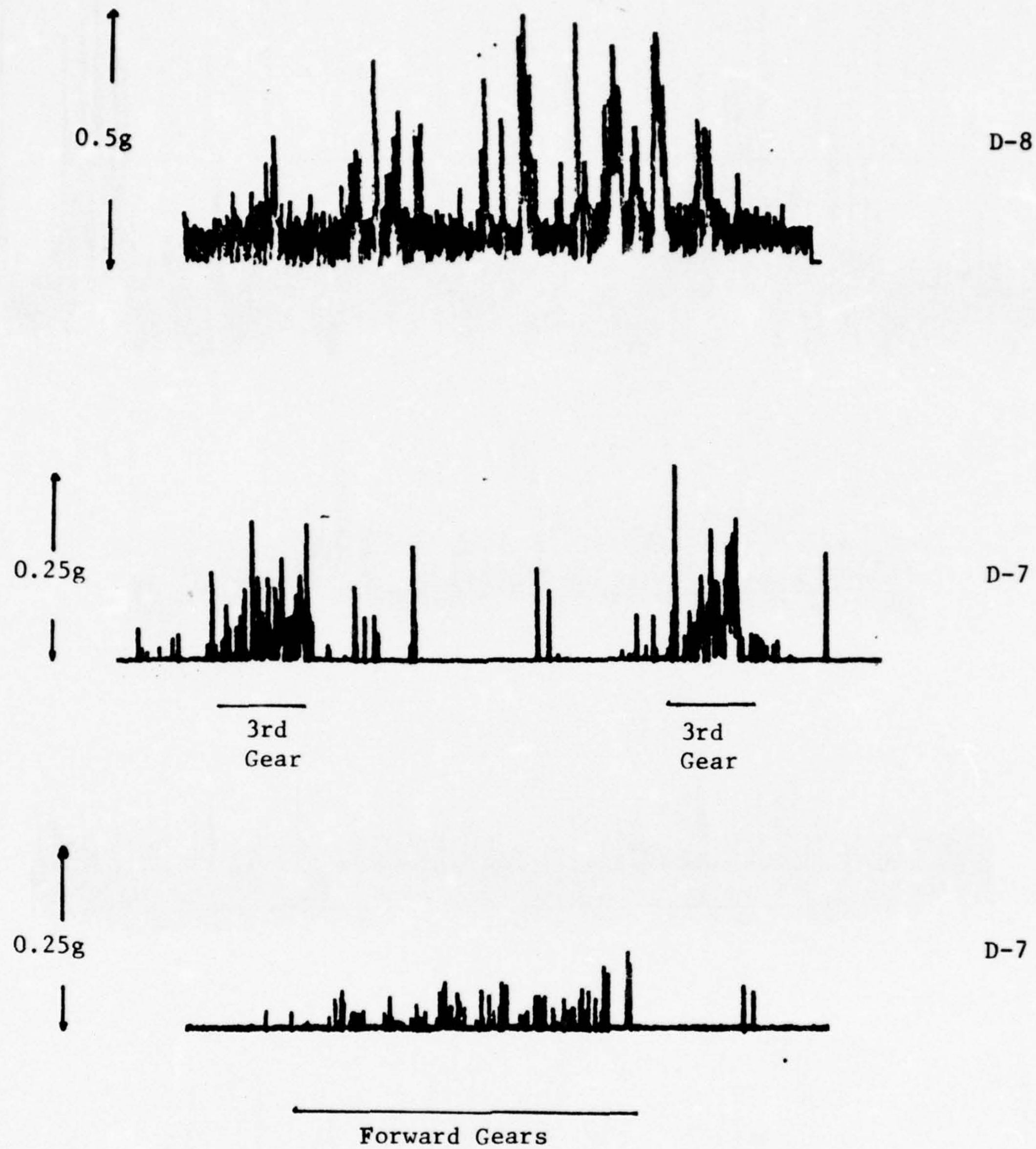


Figure 16 Final Drive Trace D7E and D8H

Figure 16 is a comparison of data taken on the final drive. The top trace is D8 data taken on the earlier program for a tractor that had a suspected minor defect in the final drive portion. The D7 data in the middle trace shows an increased amplitude in the third gear ratio. This is the same characteristic as was shown in the torque divider and is not considered significant. No effort was made in the program to run detailed analysis of various accelerometer locations to identify the source of this condition. The filtered data for the transmission does not exhibit this characteristic.

It was concluded therefore that the same basic hypothesis of the data characteristic for defective components applies to both D8's and D7's with the restriction that only the spectral region between 40 KHz and 50 KHz be utilized. Additionally, for the torque divider and for the final drive, higher levels are seen for third gear operation which should be ignored in making the diagnostic decision.

SYSTEM DESIGN

Figure 8 showed the data reduction system used to process the data from the tractors. The basic intent of the diagnostic system design was to develop an instrument that would produce comparably analyzed data on board the tractor while the tests were being performed. To ease the operation, the system should be battery powered and be rugged enough to withstand the operational environment of the crawler tractor.

Figure 17 shows the block diagram of the developed diagnostic system. Three B&K 4344 accelerometers are mounted on the torque converter, transmission and final drive respectively. These accelerometers are specifically selected to have an unmounted resonance in excess of 100 KHz. The accelerometers are connected to three charge amplifiers which convert the charge variation of the accelerometer proportional to acceleration to a voltage output. Depending on the test to be performed, one output of the charge amplifier is switched into the band pass filter. (All three outputs are available on the front panel if an external tape recorder is used.) The band pass filter passes the spectral

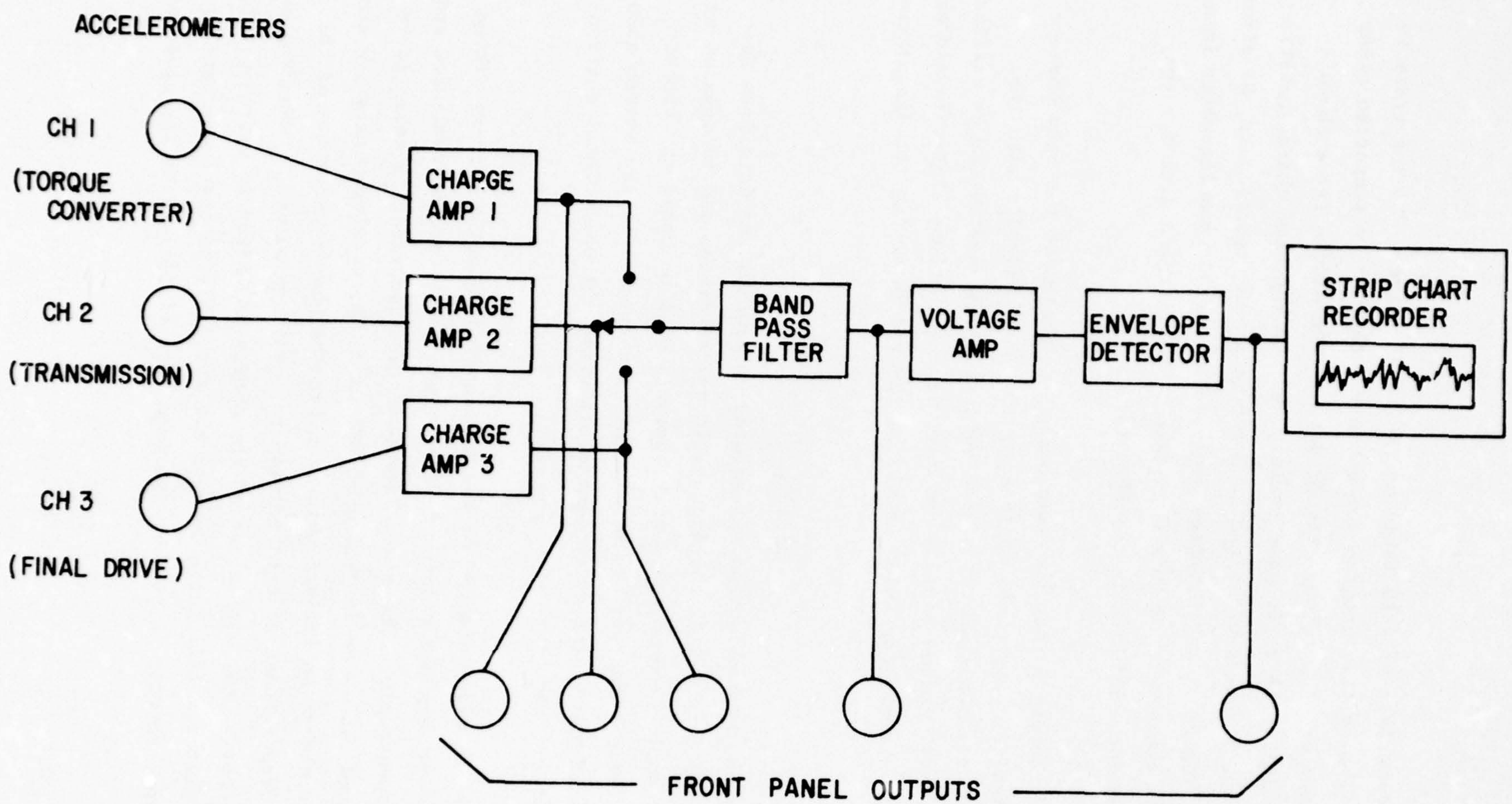


Figure 17 Block Diagram - Off-Highway Vehicle Diagnostic System

region from 40 KHz to 50 KHz. This signal is then amplified by the voltage amplifier. The envelope detector has a slow discharge time and has an output that will follow the envelope of the vibration and will hold peaks long enough to allow the strip chart to respond to fast shift transients. Not shown in Figure 17 is the power supply and the battery charging circuitry.

The Operation and Maintenance Manual furnished as Data Item 006 contains detail circuit and wiring diagrams to allow calibration and maintenance of the system. Specifications of the individual circuit cards are given below.

Charge Amplifier Card

General - The charge amplifier has two sections, a charge amplifier which converts charge to volts and a voltage amplifier for gain and sensitivity calibration specifications.

Charge Input Range	1 - 10 picocoulombs
Voltage Output	0 - 10V
Sensitivity	0 - 10 picocoulombs/unit
Output Impedance	100 ohms
Input Time Constant	8.2 ms (19.4 Hz)
Power Supply	± 15V DC ± 1% at 12 ma

Voltage Amplifier Card

General Design - The card consists of two stages of non-inverting amplifiers, each stage being gain selectable by externally grounding resistors. For this purpose the gain is fixed at 10. In addition a section is provided for overload detection and driving the overload LED.

Specifications

Input Impedance	100K ohms
Output Impedance	100 ohms
Voltage Gain	10
Frequency Response	2 Hz to 100 KHz
Output Voltage	20V p-p
Power Supply	± 15 VDC + 1% at 20 ma

Envelope Detector Card

General Description - The envelope detector has five sections. They are 1) input buffer, 2) Bipolar precision diode detector, 3) differential amplifier, 4) peak detector with time constant control, and 5) low pass filter.

Specifications

Input Impedance	100K ohms
Output Impedance	< 50 ohms
Frequency Response - Carrier	DC -100 KHz
Frequency Response - Envelope	0.1 sec.
Voltage Output Swing	0-10V
Power Supply	±15V DC ± 10% at 40 ma

Chart Recorder

The chart recorder is a commercially available strip chart recorder purchased from General Scanning Incorporated, Watertown, Massachusetts. The system is slightly modified to allow the front controls of the recorder to control the complete electronic system. The specifications are given below.

General

Number of channels	1 analog 1 event
Channel width	50mm/50 divisions
Trace presentation	Rectilinear, thermal
Chart speeds	5 and 25mm/sec. standard
Operating temperature, overall	0 to +50°C
Storage temperature	-40 to ± 60°C
Dimension	3.86H x 8.50 W x 11.35D inches
Weight	13 pounds

Electrical

Measurement range	10mv/mm to 10 volts full scale
Input circuit	1 megohm differential
Common mode rejection	80 db
Maximum signal input	± 12 volts
Non-linearity	± 1.0% of full scale
Frequency response	50mm: dc to 25 Hz 1 db down dc to 30 Hz 3 db down 10mm: dc to 90 Hz 1 db down dc to 120 Hz 3 db down

Electrical (continued)

Power	12 vdc
Power consumption	5 watts typical, 10 watts max.
Charging power	105 to 125 VAC, 60 Hz
Battery life	10 hours typical

OPERATION AND TYPICAL RESULTS

Installation

Accelerometers are mounted to the three locations on the tractor power train. Details on the mounting procedure are given in the operation manual furnished with the system.

The three accelerometer locations used are shown in the cutaway view of the crawler tractor Figure 18. Position and appearance of the D7 and D8 are similar so that no distinction is made between the two models. The accelerometer locations are (1) the top of the torque divider, (2) the right-hand forward side of the transmission and (3) the rear center of the tractor.

In order to gain access to locations 1 and 2, the right hand and center forward floor plates are removed.

Figure 19 shows the location of accelerometer 1 and 2. The view is straight down on the area with the torque divider at the top and the transmission at the bottom. The third accelerometer shown was an experimental location used during the contract work.

Figure 20 shows the close up of location 1. Figure 21 shows the close up of location 2. Figure 22 shows the location of accelerometer 3. The second accelerometer shown in Figure 22 is another experimental location.

The system is shown in Figure 23 and consists of three parts. The aluminim instrument case contains the electronics which was described earlier. The two aluminum plates at the base of the case contain shock mounts to allow the electronics to operate reliably in the high vibration level. The cast iron portion with the U bolts is an adapter to mount the system to one of the vertical pasts supporting the canopy.

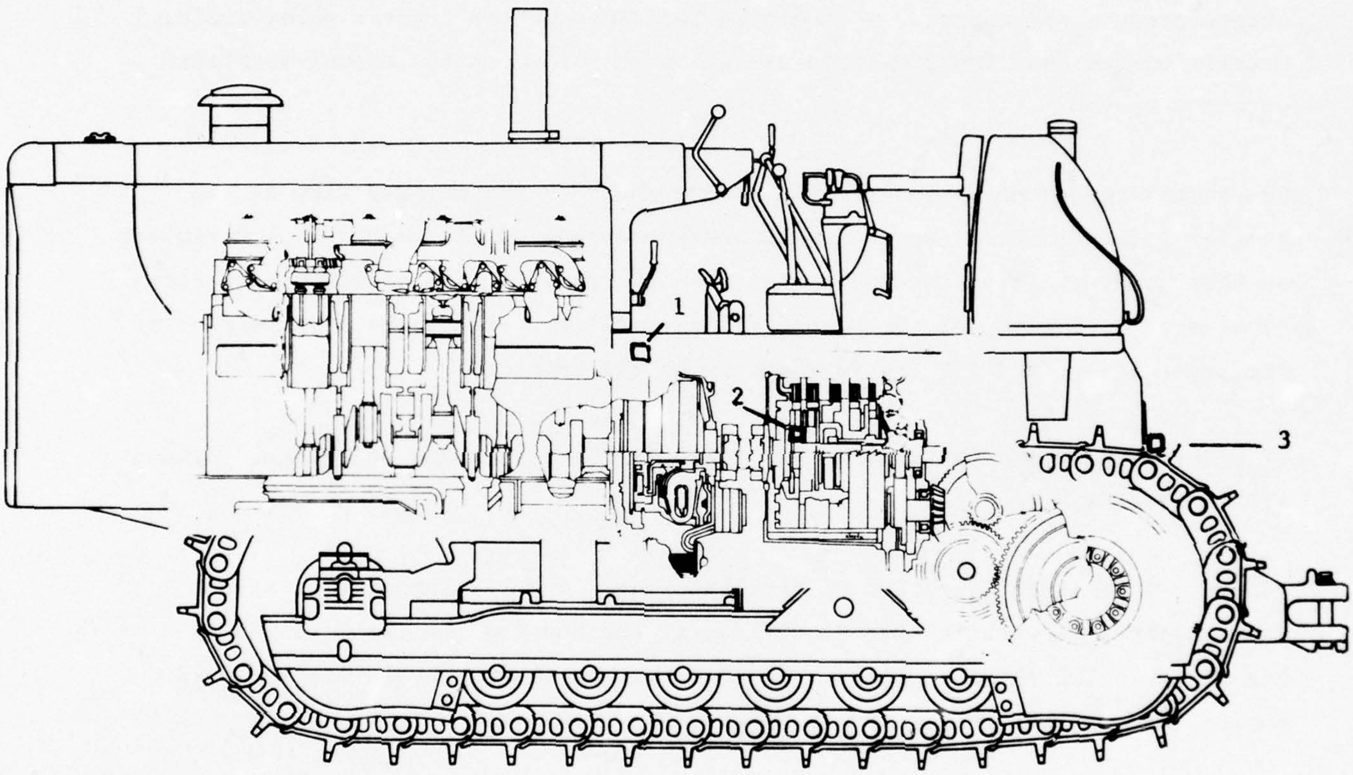


Figure 18 Accelerometer Location - Tractor Cutaway

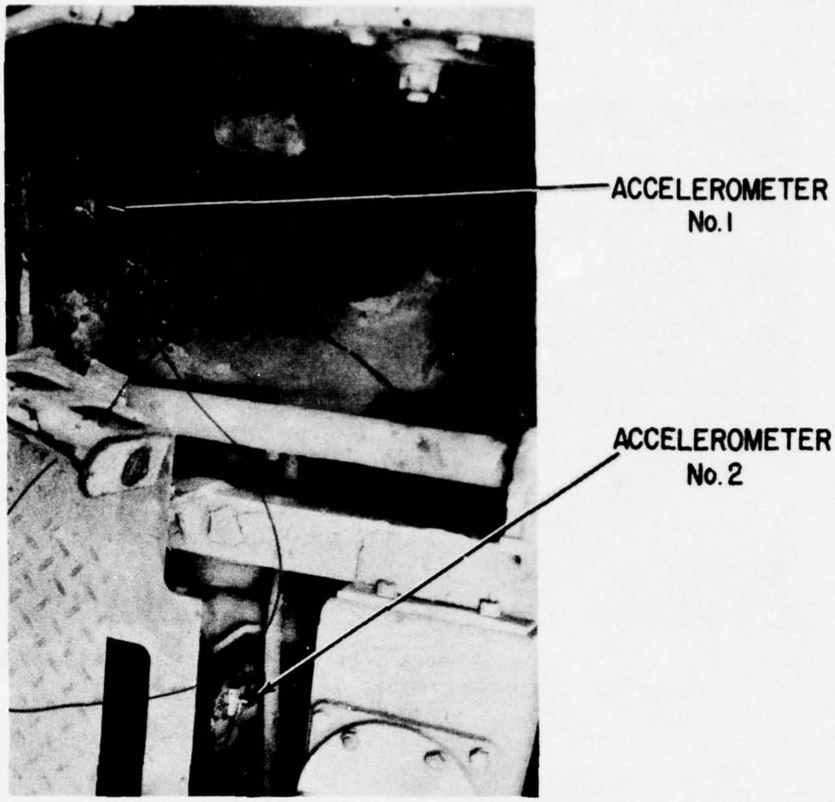


Figure 19 Accelerometer Locations No. 1 and 2

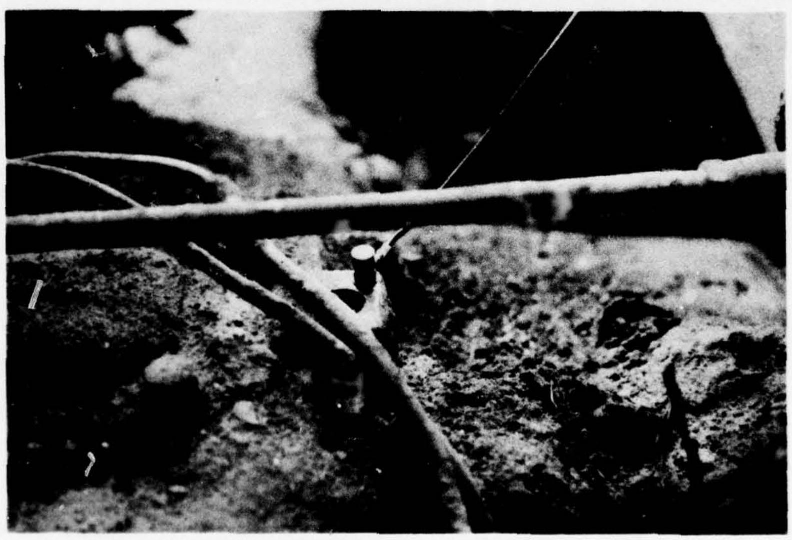


Figure 20 Accelerometer Location No. 1

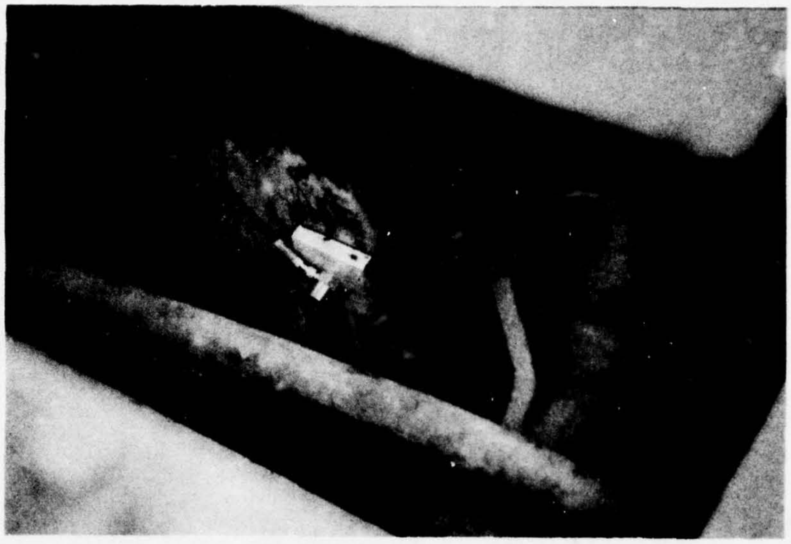


Figure 21 Accelerometer Location No. 2

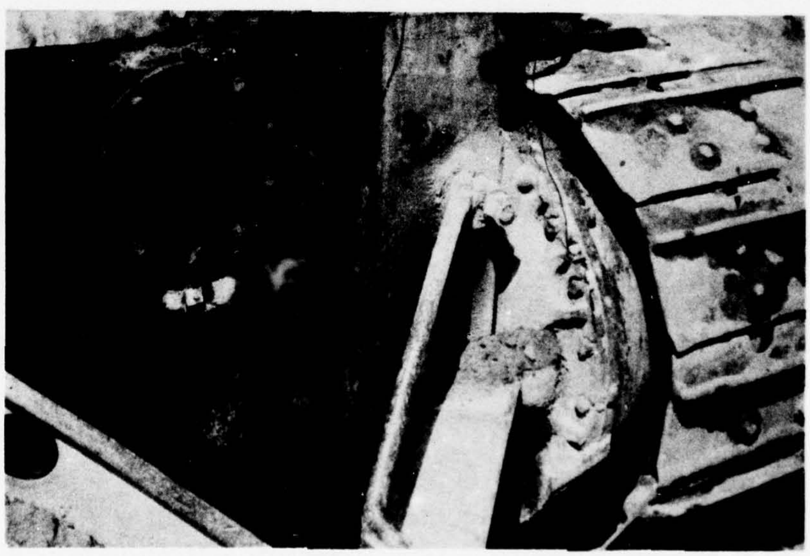


Figure 22 Accelerometer Location No. 3

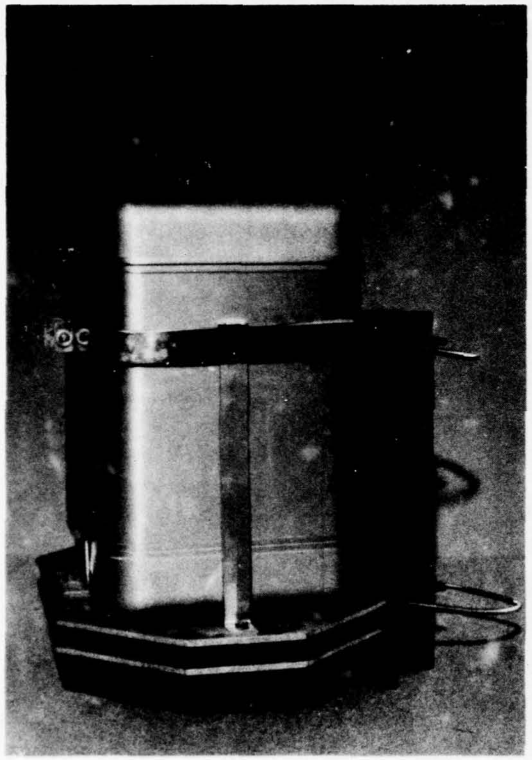


Figure 23   Diagnostic System

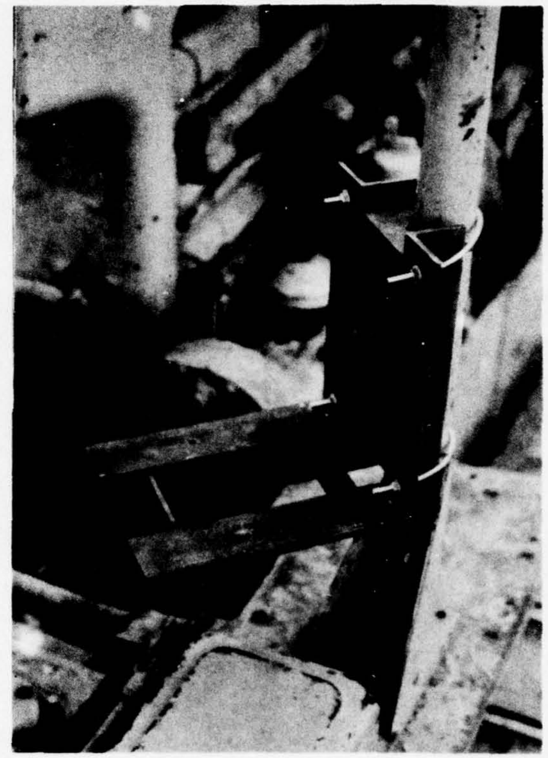


Figure 24   Mounting the Adapter

The sequence of steps in mounting the instrument is shown in Figure 24 - 26. Figure 24 shows the adapter mounted to the right-hand forward past. The shock absorber base is then bolted to the adapter as shown in Figure 25. If a tractor does not have vertical posts, this shock mount base may be bolted to the floor plates by removing an existing bolt and attaching the base to the floor plate much as is shown in Figure 25. Next the instrument case is placed on the base and the restraining band tightened as shown in Figure 26.

The lead from accelerometer locations #1 connected to Channel 1 input. Accelerometers 2 and 3 are also connected to their respective inputs. The event marker switch is taped to the shift selector and its lead connected to the EM input on the system. The system completely installed is shown in Figure 27. The operator's left hand is on the shift lever with the event marker switch installed.

Operational Sequence

Three tests are run to check the torque divider, the transmission and the final drive components. Prior to starting the first test sequence, the gain of the recorder is set to 50 mv/mm (millivolts per milimeter) and the chart speed is set to low. Setting the chart speed to low turns the whole system on. The following three tests are then run:

Test Sequence #1 - Torque Divider

Select channel #1. With the clutches engaged and the feet firmly on the brakes to prevent tractor movement go through the following shift sequence. 1st forward, 2F, 3F, 2F, 1F, neutral, 1R, 2R, 3R, 2R, 1R and neutral. Remain in each shift position for approximately five seconds. Prior to each shift, depress the event marker switch a number of times equal to the shift position that will be next selected.

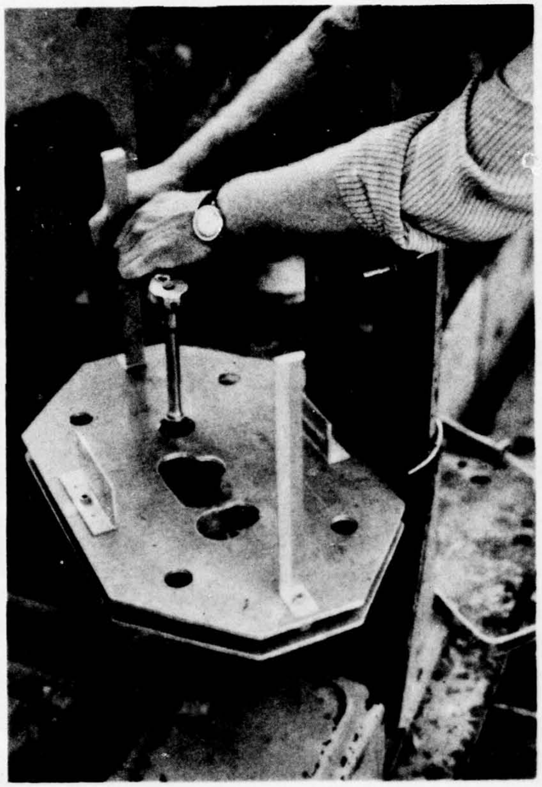


Figure 25 Mounting Shock Absorber Base

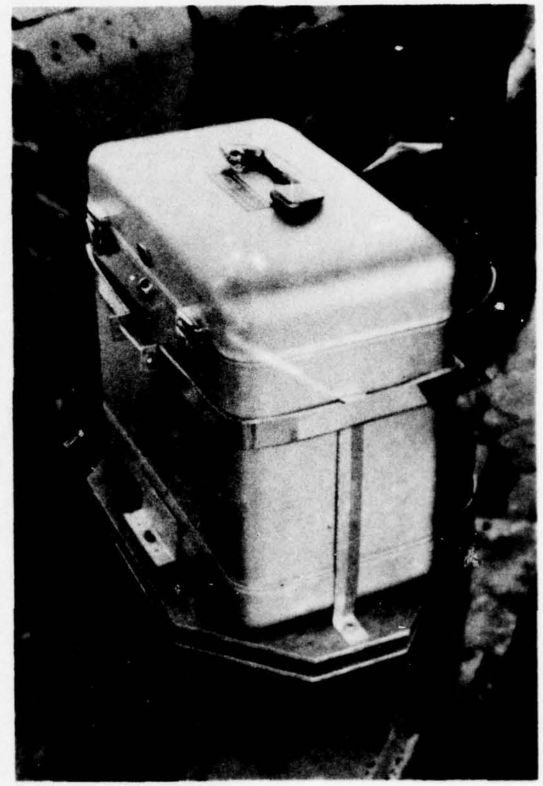


Figure 26 Mounting the Instrument Case

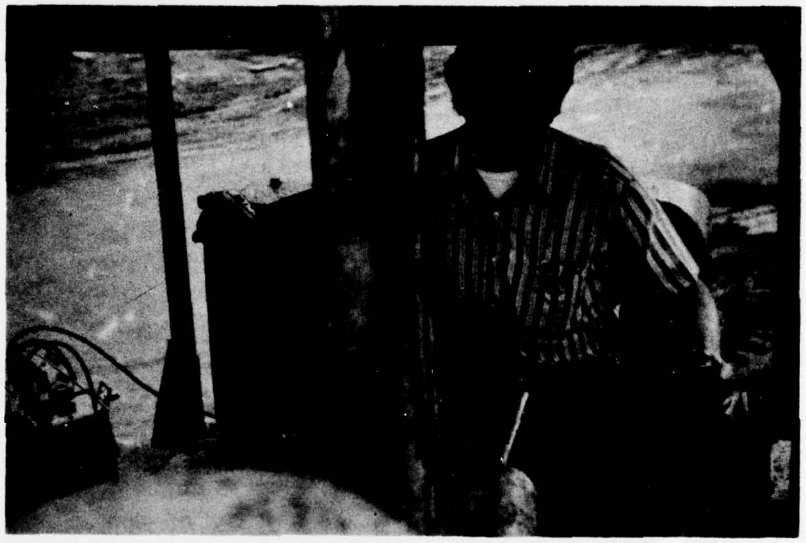


Figure 27 System Installed and Operational

Test Sequence #2 - Transmission

Select channel #2. Pull both clutches and go through the same shift sequence as in Test #1. Again depress the event marker switch to identify the next gear to be selected.

Test Sequence #3 - Final Drive

Select channel #3. Set the recorder gain to 10 mv/mm. This test sequence consists of three parts.

A. Drive

With the clutches engaged drive the tractor ahead and in reverse through the first two speed ranges. The sequence is neutral, 1F, 2F, 1F, neutral, 1R, 2R, 1R and neutral. Allow five seconds in each range as before and identify the next speed range to be selected with the event marker as before.

B. Left Clutch Pulled - Spin

Pull the left clutch and apply the left brake. Repeat the above sequence. This will produce spins to the left in forward and to the right in reverse.

C. Right Clutch Pulled - Spin

Pull the right clutch and apply the right brake. Repeat the above sequence. This will produce spins to the right in forward and to the left in reverse.

This concludes the test sequence - shut the recorder off and remove the chart paper. Label the test sequences 1, 2, 3A, 3B and 3C.

Typical Results

System outputs for a normal tractor are shown in Figures 28, 29 and 30.

Torque Divider - Figure 28

The trace is low level with minor peaks. Shift transient are low due to the nature of the torque divider. Abnormal torque dividers would produce a higher average level with peaks to the chart full scale.

Transmission - Figure 29

The shift transient appears as a peak on chart paper. In the normal transmission shown, each shift transient is followed by a low level signal. For an abnormal tractor, the region between the shift transient would be equal to or higher than the shift transient.

Final Drive - Figure 30

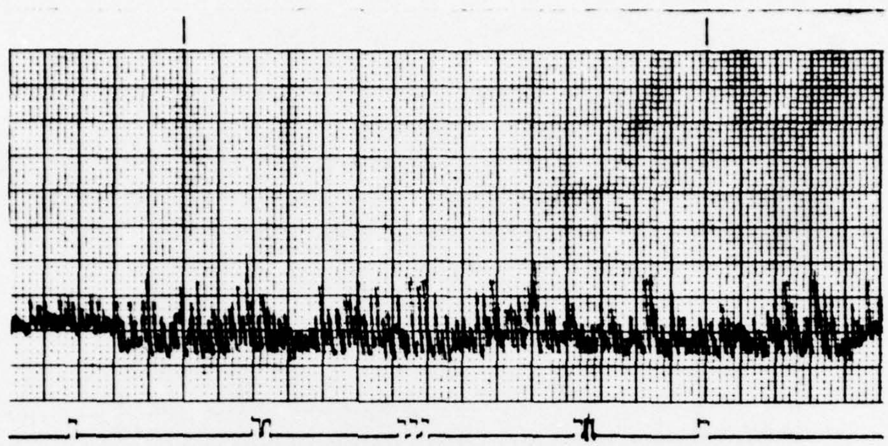
Only the chart for the drive test is shown. Third gear was used, however this is not necessary. The differences between normal and abnormal times is the same as for the transmission.

An example of a faulty transmission is shown in Figure 31. This trace was produced by playing the tape recorded data from a possibly defective D8 transmission recorded in the previous program into the electronic system after the charge amplifier. Gains were adjusted so that the representation is what would have been obtained if the test had been run real time.

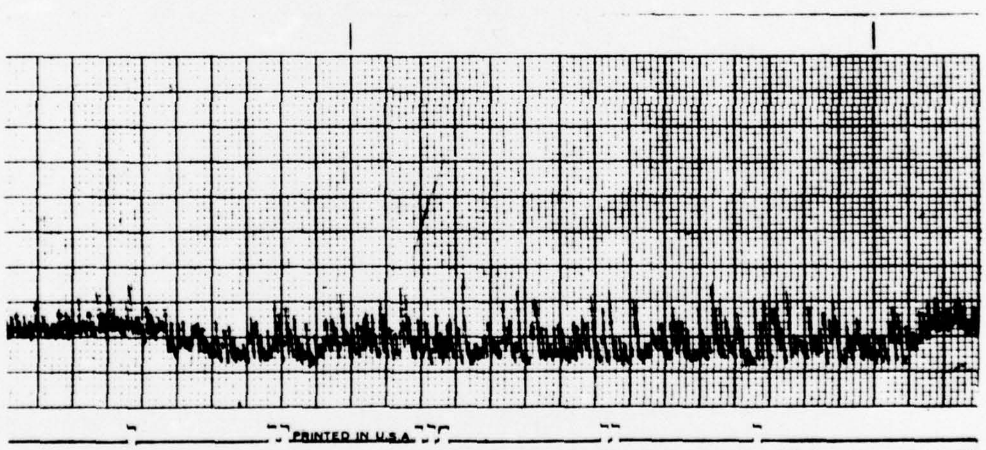
Diagnostic Logic

Torque Divider

The maximum load on the torque divider exists in the stall test. The operation of the torque divider is the same regardless of what shift position is selected. Therefore, the torque divider is suspect if the levels are high in <u>all</u> shift positions. The full shift sequence is used to rule out some unidentified transient effecting the data in only one test.



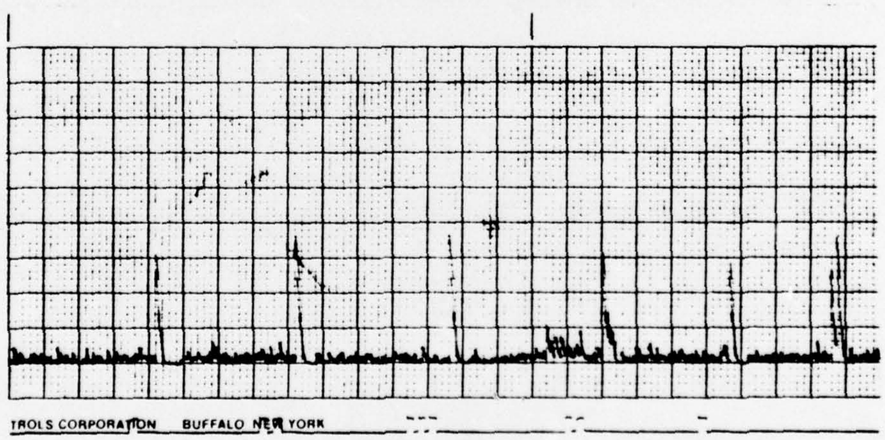
a) Forward



b) Reverse

Figure 28 Typical System Output - Torque Divider

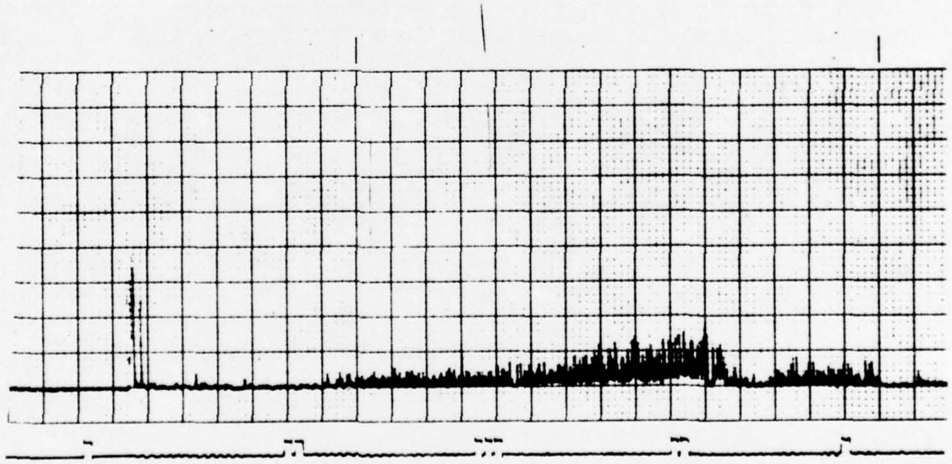
a) Forward



b) Reverse

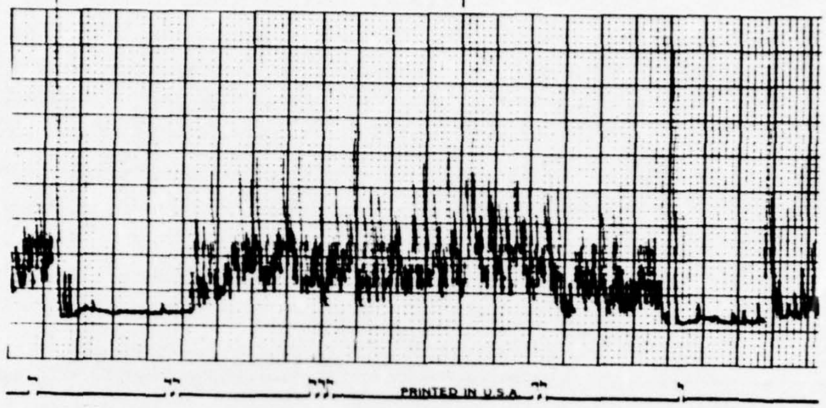
Figure 29 Typical System Output - Transmission

a) Forward

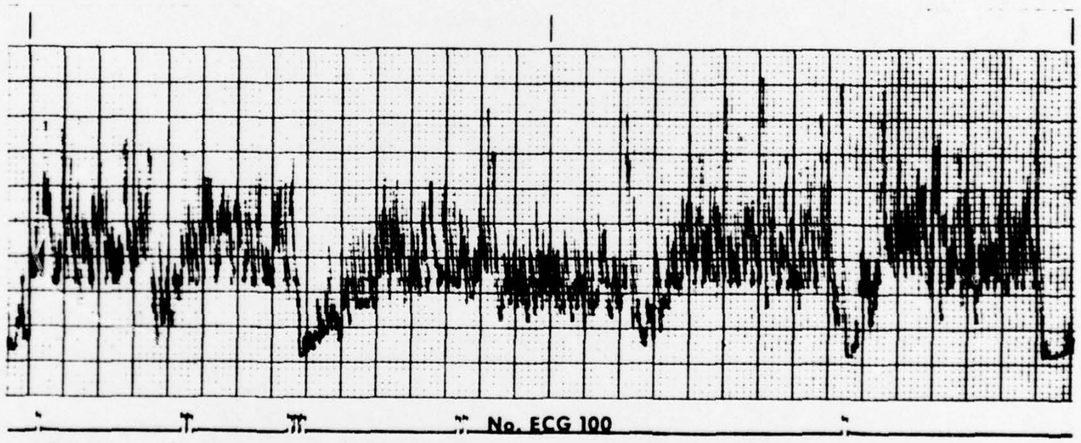


b) Reverse

Figure 30 Typical System Output - **Final Drive**



a) Forward



b) Reverse

Figure 31 Typical Faulty Transmission Output

Transmission

Only certain positions of the transmission are loaded in each shift position. It is therefore possible to develop a logic to identify the component of the transmission that is faulty based on the portion of the data trace that is high. If for example only the forward planetary is faulty, then the high level signals will be present in all the forward gears and in none of the reverse. Figure 32 is a fault matrix table that contains the logic. If the D8 data is analyzed from Figure 31 the following conclusions may be drawn:

1. The problem does not exist in the forward planetary or in the reduction gear since high level signals do not exist in first gear forward.
2. The only explanation for the high level signals in first reverse and not first forward is that the reverse planetary is faulty.
3. High level signals in second and third forward would be due to faulty #2 and #3 planetaries.

Final Drive

Defective components in the right and left hand final drive portions may be determined if high level signals exist in only one series of spins i.e. when one clutch is pulled and one brake applied. If high level signals exist in both series, the most likely cause would be the bevel reduction gear.

It must be emphasized that the above logic must be verified in the future since none of the tractors analyzed in this or the previous program were available for teardown and inspection.

Fault Matrix

Symptom Present in Gears	#1 Forward	#2 Reverse	#3 (2)**	#4 (3)**	#5 (1)**	Reduction Gear
1F	X		X*		X	X
2F	X		X			X
3F	X		X*	X		X
1R		X	X*		X	X
2R		X	X			X
3R		X	X*	X		X

* May be differentiated by lower levels due to shared load.

** Train number counting toward rear.

Figure 32 Symptom Fault Matrix - Transmission

CONCLUSIONS

1. D7 crawler tractor data recorded and analyzed during this contract differed from earlier D8 data in the region from 5 KHz to 35 KHz. In the case of the D7, high level signals were present in this spectral region for steady-state operation in third gear forward and reverse. This was not the case for D8 data.

2. By restricting the spectral region of interest to between 40 KHz and 50 KHz, data from both the D7 and D8 crawler tractors are essentially the same during the shift transient and during steady-state operation.

3. The diagnostic hypothesis developed in previous work for the D8 therefore seems applicable to the D7 as well.

4. An engineering model of a diagnostic system was designed, fabricated and tested on Caterpillar crawler tractors. In an operational demonstration, a complete diagnostic test, including installation of the system, tractor testing and system removal was conducted in less than one hour.

5. The diagnostic system developed, along with the diagnostic logic, allows a rapid assessment of a tractor's condition by an engineer or technician in the field.

6. While the system operates as designed and the diagnostic logic seems consistent with the signals produced by defective components, this concept has not been proven by carefully controlled tests in which component condition is carefully correlated with the signals produced and the diagnostic logics hypothesized.

RECOMMENDATION

Now that the system has been developed there is one major recommendation. The system, based on a diagnostic hypothesis, must be tested. In order to be evaluated the system should be operated on a series of D7 and D8 crawler tractors that are scheduled for teardown and repair. The condition of the power train components must be predicted based on system outputs prior to teardown. The conditions of the power train components must be determined during teardown and these results checked against the prediction.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

An earlier report prepared under Contract DAAK02-74-C-0084 investigated the feasibility of utilizing high frequency vibration analysis to diagnose power train condition in off-highway vehicles. The program was conducted on Caterpillar D8H crawler tractors and the results were promising. The present program investigates the commonality of the data from a D8 and a smaller crawler tractor the D7. → next page

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cont.

Based on the data from the D8, a spectral region between 20 KHz and 50 KHz was selected. This wide band width did not work for the D7 due to the presence of high level signals during steady-state operation in the 5 KHz to 35 KHz region. It was found that if the band width was limited to 40 KHz to 50 KHz, the diagnostic technique would apply equally well to both the D7 and D8 tractor.

An engineering model of a diagnostic system was built and tested that will operate on both the Caterpillar D7 and D8 crawler tractors.